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ACTIVE STRUCTURAL ACOUSTIC CONTROL

AND

SMART STRUCTURES

FINAL TECHNICAL REPORT

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ABSTRACT

This final technical report for the DARPA/ONR Advanced Submarine Technology (AST) Program concerns recent VPI&SU research in the area of active structural acoustic control with active/adaptive structures. The material is divided into progress in the areas of structural acoustics, actuators, sensors, and control approaches. Due to the coupled nature of the problem, considerable effort throughout the VPI&SU program has been given to the interaction of these areas with each other, and their integration. The focus is analytical and experimental investigations of active control of sound radiation from simple panels as well as panels with some added complexity using point and distributed control forces and sensors on the plates. The results presented show that significant progress has been made towards controlling structurally radiated noise by active/adaptive means applied directly to the structure. The demonstrated technology is highly applicable to acoustic quieting in advanced submarine designs.

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EXECUTIVE SUMMARY

This research for the DARPA/ONR Advanced Submarine Technology (AST) Program in the area of active structural acoustic control with active/adaptive structures has spanned three years of effort at VPI&SU. The results show that significant progress has been made in the areas of structural acoustics, actuators, sensors, and control approaches. Due to the coupled nature of the problem, considerable effort throughout the VPI&SU program has been given to the interaction of these areas with each other, and their integration. The focus is analytical and experimental investigations of active control of sound radiation from simple panels as well as panels with some added complexity using point and distributed control forces and sensors on the plates. Structurally radiated noise controlled by active/adaptive means applied directly to the structure should be highly applicable to acoustic quieting in advanced submarine designs.

The key technical accomplishments under this Grant have advanced the state-of-the-art in structural acoustics, adaptive materials and structures, and active control. Concisely some of the important accomplishments are:

1. Demonstrated that structural acoustic control has a number of marked advantages over the use of acoustic secondary sources and sensors.
2. Applied active structural acoustic control to beam, plate and cylindrical structures.
3. Demonstrated active structural acoustic control of narrowband, wideband, and transient disturbances.
4. Developed, implemented and applied multi-input multi-output (MIMO) adaptive LMS feedforward and closed loop state-space control theory designs.
5. Optimal active structural acoustic control approaches represent a significant reduction in the dimensionality of the control system compared to that of conventional active structural vibrational control.
6. Developed design approaches and algorithms for the optimal placement of point force and distributed PZT actuators for actively controlling acoustic radiation from plates.
7. Similarly, developed the design approaches for the optimal placement of point accelerometer, and distributed (attached and embedded) PVDF sensors.
8. Demonstration of optical fiber sensing for structural acoustic control of radiation.
9. Developed an eigen-analysis approach for designing shaped PVDF distributed structural acoustic sensors
10. Extended the capability of the DTRC NASHUA numerical structural acoustic model to address active control approaches for fluid loaded structures.

11. Analytical and experimental demonstrations and investigations of acoustic wavenumber control.
12. Demonstrated state-space control theory approaches which incorporate observers that estimate the acoustic farfield from structural measurements.
13. Development of a transputer-based parallel processing controller hardware system for multi-input multi-output (MIMO) experiments.
14. Demonstrations of embedded Shape Memory Alloy (SMA) fibers in composites for adaptive structural tuning to reduce acoustic radiation.
15. Implementation of embedded PVDF sensors and theoretical analysis of embedded PZT actuators in composites.
16. Extended basic feedforward adaptive control approaches and theory through an examination of the eigen-structure.
17. Technology transfer to defense industries for further development into state-of-the-art applications, and products.

The results of this research have been presented in 127 technical papers (including papers in refereed journals and those submitted), and 20 talks at symposia and conferences.

SECTION 1.0

INTRODUCTION

This is the final technical report for Office of Naval Research (ONR) Grant N00014-88-K-0721 as part of the Defense Advanced Research Project (DARPA) Advanced Submarine Technology (AST) Program. It addresses Virginia Polytechnic Institute and State University (VPI&SU) research in the area of active structural acoustic control with active/adaptive structures. The report is divided into progress in the areas of structural acoustics, actuators, sensors, and control approaches. Due to the coupled nature of the problem, considerable effort throughout the VPI&SU program has been given to the interaction of these areas with each other, and their integration into active structural acoustic systems. The results presented show that significant progress has been made towards controlling structurally radiated noise by active/adaptive means applied directly to the structure. These technologies are important to advanced submarine designs in meeting future quieting requirements.

1.1 STATEMENT OF WORK

A concise statement of work for this research program is:

"Analytically and experimentally investigate various methods, control strategies, and controller designs for active structural acoustic control. This shall be a multi-discipline effort using personnel with significant experience in adaptive acoustic control, active methodology, and distributed actuator and sensor technology. The work will focus on controlling sound radiation from simple and complex panels with various structural discontinuities. The control will be implemented by discrete active point forces, by piezoelectric distributed actuators, and shape memory alloy (SMA) reinforced composite."

A number of goals and milestones were set out at the beginning of this research effort, and they are concisely summarized in Figure 1 for the three year research effort. The research task schedule was divided into analytical and experimental efforts. The period of performance for this grant was from 1 September 1988 to 31 August 1991. The Scientific Program Officer was Dr. Albert J. Tucker, ONR Mechanics Division.

1.2 PROGRAM ORGANIZATION

This section will present the organization of this research program at VPI&SU, and discuss cooperative programs associated. Professor Chris R. Fuller is the Principal Investigator for this research program, and he is assisted by a number of other faculty co-investigators and staff members as shown in the organization chart illustrated in Figure 2. This chart shows the top level structure of the program organization, and the primary faculty and staff members with their respective functions. The key participants are in the Mechanical Engineering Department at

Year 1

Analytical

- Analytical model for adaptive control of sound radiation from panels by input forces
- Modelling of distributed SMA actuators, piezoelectric elements, and optical sensors

Experimental

- Construction of experimental rig
- Hardware implementation of control algorithms
- Fabrication of optical sensors and integrated distributed actuators
- Experimentation on active control of freefield sound radiation from simple panels

Year 2

- Extend analytical model to include added complexities - ribs, thickness change, etc.
- Develop optimal control laws for SMA and piezoelectric actuators
- Addition of non-linear considerations to SMA actuator models
- Develop Artificial Intelligence concepts for control parameter estimation

- Demonstrate closed loop control with SMA actuators and optical sensors
- Experiments on control of free-field sound radiation of complex panels
- Testing of active control of panel radiation in a reverberant environment

Year 3

- Analysis of control of sound radiation by adaptive modal modification and strain energy tuning
- Develop a simulation of optimal control laws for SMA actuators, piezoelectric elements, and optic sensors- high-order

- Experiments on control of sound radiation by modal modification and strain energy tuning
- Demonstrate higher order input control of SMA actuators with AI concepts
- Optimize advanced LMS control and control parameter choices

Figure 1 Schedule and Milestones

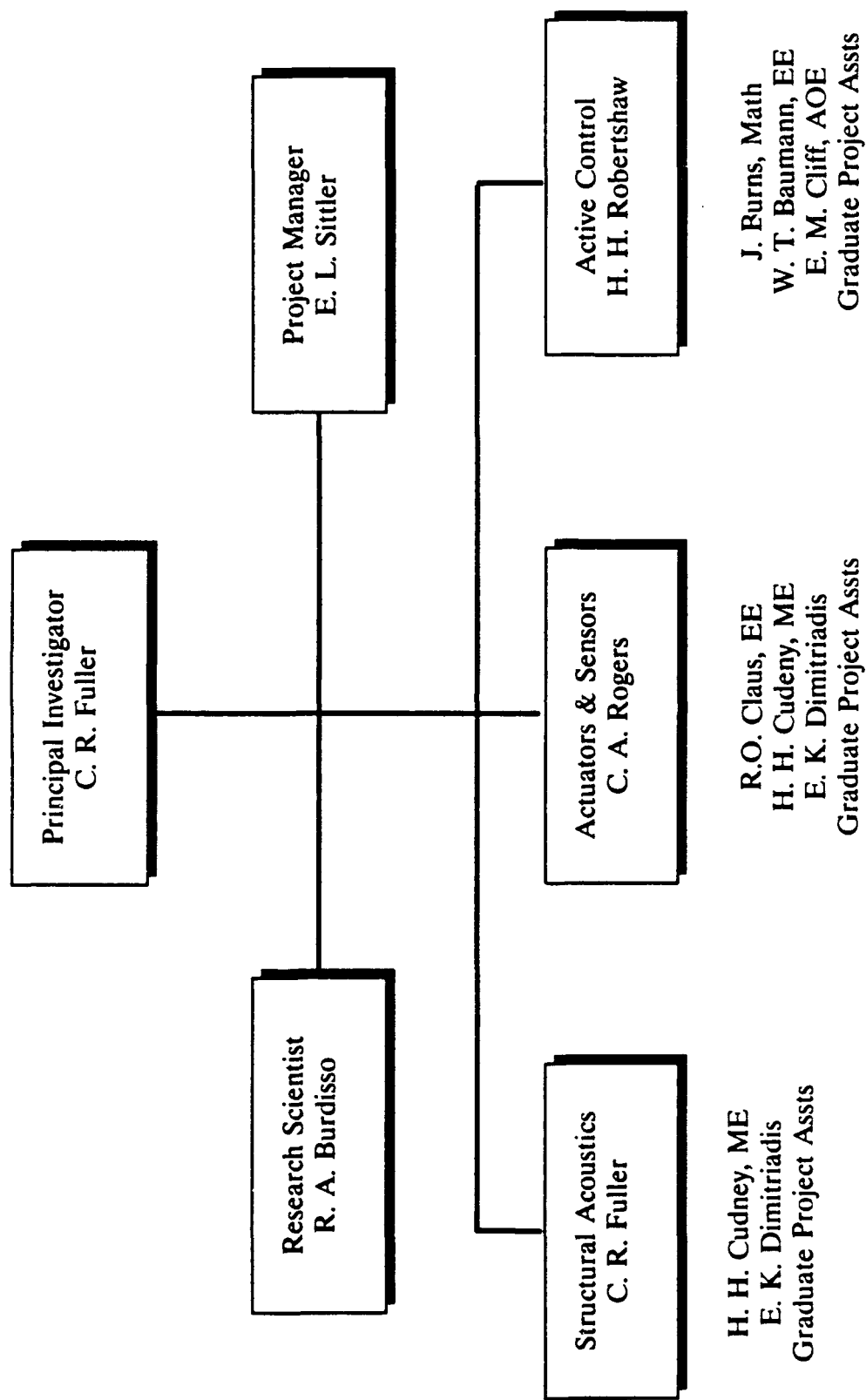


Figure 2 Program Organization

VPI&SU, and they are assisted by members of the Electrical Engineering, Aerospace and Ocean Engineering, and Mathematics Departments as shown. The overall structure of this organization is based upon the critical technology for active structural acoustic control, and the individual assignments reflect well established expertise in the respective technologies and functional requirements. A tabulation of all the VPI&SU participants is provided in Appendix A which additionally lists students and their degree programs.

Cooperative Research Programs

A goal in this project has been cooperation with centers of excellence which are performing related research work. Participation of staff from centers at VPI&SU was with the view of widening the base and strength of the program. This has involved the Center for Intelligent Material Systems and Structures, Fiber & Electro-Optics Research Center, and The Interdisciplinary Center for Applied Mathematics.

With the formal approval of ONR and other organizations there has been a very productive and cooperative effort with other research programs outside VPI&SU that involve similar technical interests and complimentary resources. In brief, these cooperative programs are:

- NASA Langley Research Center- Exchange of LMS control methodologies and experimental implementation.
- Naval Research Laboratory (NRL)- Active control experiments performed on-site at NRL acoustic holography facility. Application of optical fibers to NRL research.
- University of Adelaide, Australia- Exchange of faculty/students in areas of transputer type processors, piezoceramics actuator development.
- David Taylor Research Center (DTRC)- Development of control module for NASHUA, use of computers, loan of equipment, and exchange of information on the subject of fluid loaded structural modeling.
- University de Technologie de Compiegne, France- Exchange of faculty/students in areas of signal processing and control development.

1.3 REPORT ORGANIZATION

The organization of this report is to present the rationale for this research and the progress made in the important areas of structural acoustics, actuators, sensors, and control approaches. The next section (2.0) presents the technical background and rationale for active acoustics, the state-of-the-art at the beginning of this research, VPI&SU's historical involvement, and the specific research objectives and goals of this research program. Section 3.0 presents an overview of active structural acoustics and involved technologies. Sections 4, 5, and 6 address the research progress made in the areas of structural acoustics, actuators, and sensors

respectively. Both state feedback and LMS adaptive feedforward control techniques are discussed in Section 7. Section 8 addresses the application and transfer of these developed technologies to practical submarine quieting problem areas, and advanced designs. Section 9 presents the future directions of the continuing research in terms of the final goals and perceived problems at the conclusion of the reported research. Section 10 gives a summary and the conclusions that can be drawn from the research undertaken. The last section (Section 11) is the reference list for this final technical report. Appendices are included to show project participants, and a bibliography of technical papers and talks supported by this research Grant.

SECTION 2.0

BACKGROUND

In many applications the sound fields radiated by vibrating elastic structures is an important noise problem. Examples are machinery noise in factories, marine hull radiated noise and interior noise in aerospace applications. Of particular interest is submarine structural acoustic radiation where there is a requirement for extreme levels of quietness, and it additionally represents a situation with seawater fluid loading and propagation. An understanding of the behavior of such systems, with a view to controlling them, involves the field of structural acoustics. Structural acoustics is the study of how elastic structures radiate or receive sound [1] and in its most fundamental form involves the simultaneous solution of the differential equations describing the structure and fluid media with appropriate coupling conditions between the two (a "fully coupled" analysis).

The usual methods of control of such radiated noise fields involve passive techniques such as added mass, damping, stiffness and system modification through re-design. However, these techniques have proved unsatisfactory in many applications for a number of reasons. Passive techniques usually imply a significant mass increase, do not work well at low frequencies and due to the "coupled" nature of the problem it is difficult to predict their effect except in the simplest of applications. For submarines the very low frequency radiated noise signature has gained increased importance, and passive technologies are solidly confronted with a number of limitations. One approach that shows much potential to overcome these difficulties is active control. While active acoustic control was originally proposed in 1933 by Professor Paul Leug in Germany [2] it has only recently re-emerged as a very promising technique to reduce radiated sound fields [3]. The main reason for this has been advances in high speed data acquisition and processing enabling active control to be implemented in "real time". The most practical and frequent active acoustic implementations have been restricted to simple one dimensional air duct noise propagation problems. The traditional active approach to the problem of reducing sound radiation from structures is to use a number of secondary acoustic control sources arranged around the structural noise source (see for example Ref. 4). This technique can be seen to have a number of disadvantages [5]. From an implementation point of view, it is often impracticable to have secondary acoustic sources located away from the structure (as well as error microphones or hydrophones). More importantly, the use of acoustic control sources leads to a generation of additional unwanted noise, termed "control spillover". This situation is overcome by using additional strategically positioned acoustic sources. Thus, for example, a machine radiating noise requires a large number of active acoustic sources arranged around it to produce global (here global means throughout an extended volume or space) noise reduction [4].

VPI&SU has a long history of accomplishments in acoustics, active structural control and intelligent materials which is internationally recognized, and its engineering graduates have gone on to be leaders in these fields as well. Structural acoustics is one of the many applications of

these technologies that VPI&SU has focused upon in the past, and significant successes have been developed for the aerospace industry and NASA. As this report illustrates, VPI&SU has assembled a team of expertise to pursue the advancement of active structural acoustics for the DARPA Advanced Submarine Technology Program.

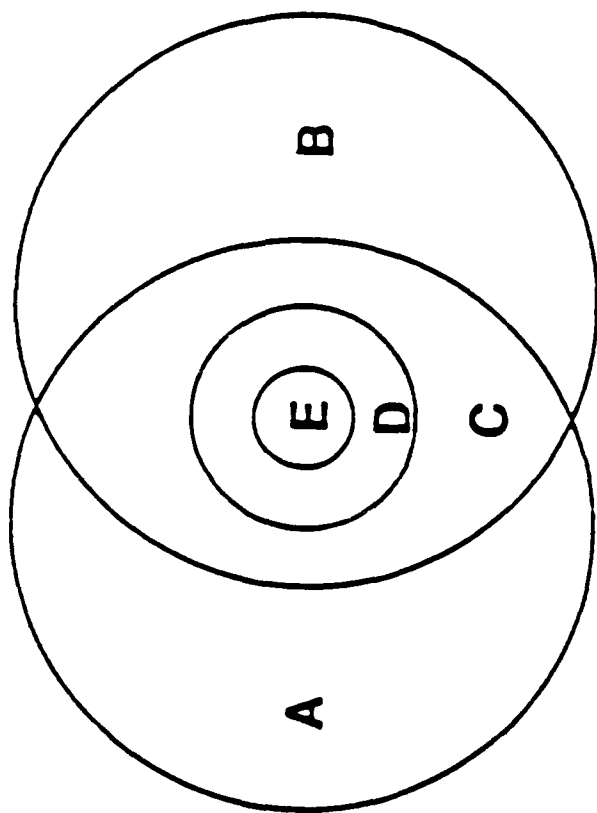
SECTION 3.0

ACTIVE STRUCTURAL ACOUSTICS OVERVIEW

The purpose of this research is related to a new technique in which the noise radiated from vibrating structures is reduced by active inputs applied directly to the structure itself. The active inputs may take the form of oscillating forces or strains, adaptive steady state forces or adaptive changes in material properties. While the control inputs are applied directly to the structure, the error information (i.e, the variable to be minimized) is usually taken from the radiated acoustic field. The subtlety of this approach is that one can take advantage of the nature of the behavior of the structural acoustic coupling in order to optimize and reduce the dimensionality of the controller. The discipline thus involves the study of the interaction of controller-structure-acoustics. A patent on the technique has been awarded to the Principal Investigator in 1987 [6]. As the aim of the work is to develop structures with embedded actuators and sensors as well as develop "intelligent" control approaches, the work falls into the domain of what is called adaptive, active, or "smart" structures.

The rapidly emerging field of adaptive structures and the early work that stimulated it, is well summarized in the review article by Wada et al [7]. Figure 3, which was based on Ref. 7, shows a general framework for adaptive structures. The two most basic systems are the sensory structures in which only information related to the states or parameters of the structure are measured. Adaptive structures are those which include actuators which can alter the system states or parameters in a controllable way. The interaction of these two areas is defined by Wada et al [7] as "controlled structures," and thus include some degree of control intelligence in dictating the required inputs to the actuators to achieve the required sensed states or parameters. A finer definition on controlled structures is "active structures" in which the actuators and sensors are highly integrated into the structure itself. Due to the fact that the sensors and actuators are directly part of the structure (bonded or embedded) one may then view the structure itself adapting its properties in an active way to a disturbance. Finally, when the control elements and electronics and perhaps even power sources are directly integrated into the structure itself, the structure is defined to be "intelligent" in that it appears to provide its own support energy and cognitive ability (although it admittedly had to be "taught" its process at one stage). For a detailed discussion of the foundation work in sensors, actuators, and applications the reader is referred to Ref. 7.

In terms of controlling sound radiation from structures using active or adaptive means applied directly to the structure while sensing some state or parameter of the structural acoustic system, very little previous work has been done. Knyasev and Tartakovskii [8] demonstrated in a brief experiment that it was possible to reduce sound radiation from a panel using an active damping force. Vylayshev et al [9] analytically considered sound transmission through an one-dimensional plate and showed an increase in the plate transmission loss with one active control force applied to the plate. Fuller and Jones [10] and Jones and Fuller [11] showed both experimentally and analytically that sound transmission into a model elastic fuselage can be



- A.....Adaptive Structures
- B.....Sensory Structures
- C.....Controlled Structures
- D.....Active Structures
- E.....Intelligent Structures

Figure 3. Framework for structures with integrated functions.

globally controlled with a small number of point force control inputs applied to the structure. This early work [10,11] defined the basis of the research in the present project; while the control inputs are applied to the structure the error information is taken from the acoustic field (or structural motions that acoustically radiate) and thus the controlled system includes the natural coupling between the structural response and the acoustic field. This approach will be shown, for low frequency applications, to markedly reduce the dimensions of the controller.

The following sections will detail VPI&SU's work on this new active structural control technique, and the technologies developed for its implementation. For clarity, the discussion is divided into the four areas; structural acoustics, actuators, sensors and control approaches. However, the research is always a complete synthesis of the above four areas due to the "coupled" nature of the problem. For brevity this report is limited to a discussion of the concepts and results; details of the experiments or analyses can be found in the appropriate references and bibliography. The authors believe that this report defines a rapidly emerging new field in noise control that has an important role in advanced submarine and other vehicle designs, aerospace, and industrial applications.

SECTION 4.0

STRUCTURAL ACOUSTICS

Structural acoustics is directly concerned with the coupling between the motions of elastic structures and their radiating (or receiving) sound fields. The response of these systems must be solved simultaneously (or in a coupled sense) as in heavy fluids or highly reactive environments where the back loading (radiation impedance) of the acoustic field affects the motion of the radiating structure [1]. Obviously this natural "feed-back" loop has important implications on the design of active systems for such situations. This section will discuss some of the important characteristics of structural acoustic coupling and how they relate to the present problem as well as VPI&SU contributions to the field.

One important aspect of how structures radiate sound is known as radiation efficiency σ , which is defined as the ratio of the acoustic power that a structure radiates to the power radiated by a piston of equivalent area vibrating with an amplitude equal to the time-spatial average of the structure. Obviously the higher σ is, the more sound energy will be radiated for a given structural response. As an example, Figure 4 shows the radiation efficiency of various modes of vibration of a baffled circular clamped plate versus non-dimensional frequency $k_a = 2\pi fa/c$, where f is frequency of excitation, a is plate radius, and c is the speed of sound [12]. For low frequencies, σ is seen to vary strongly with modal order, (m,n) where m and n are circumferential and radial mode numbers, respectively. The most efficient mode is the $(0,1)$ or monopole like mode. All modally responding structures exhibit behavior related to Figure 4. Thus, if a structure is vibrating in a number of modes of nearly equal amplitude then only a few of these will radiate sound in the low frequency region (which is our frequency region of interest). This implies that only one or two control inputs need be applied to the structure to reduce the radiated field as long as the error information is constructed from radiated acoustic variables.

Figure 5 shows results from an experiment in which sound transmitting through an elastic clamped circular panel located in a baffle is reduced by active point force inputs applied to the panel [13]. The error points are two microphones in the radiated acoustic field. In this experiment the aluminum plate was 0.4572 m in diameter and 1.27 mm thick. The incident acoustic wave was planar at an angle of incidence of 45° and was at a single frequency of 112 Hz. Substantial reduction in radiated noise is observed in this off-resonance case where there are many panel modes present in the response. In contrast, Figure 6 shows the same situation except that the error information is taken from two accelerometers mounted on the panel (i.e., directly controlling panel vibration). The results show that, although panel vibration is attenuated by around 20 dB at the error points, the noise radiation increases. This effect is termed "control spillover".

Radiation impedance, a related variable, is also an important characteristic to be considered. Figure 7 shows analytical results for a similar situation as discussed above. Here the cost

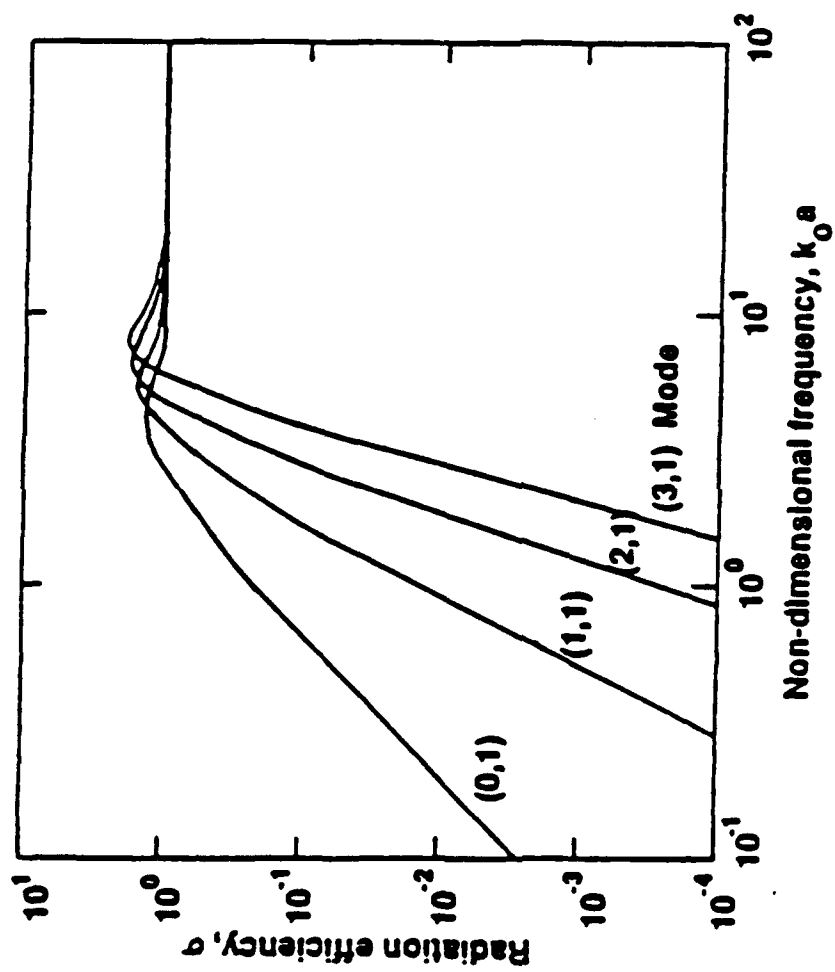


Figure 4. Modal radiation efficiency of clamped circular panel in a baffle.

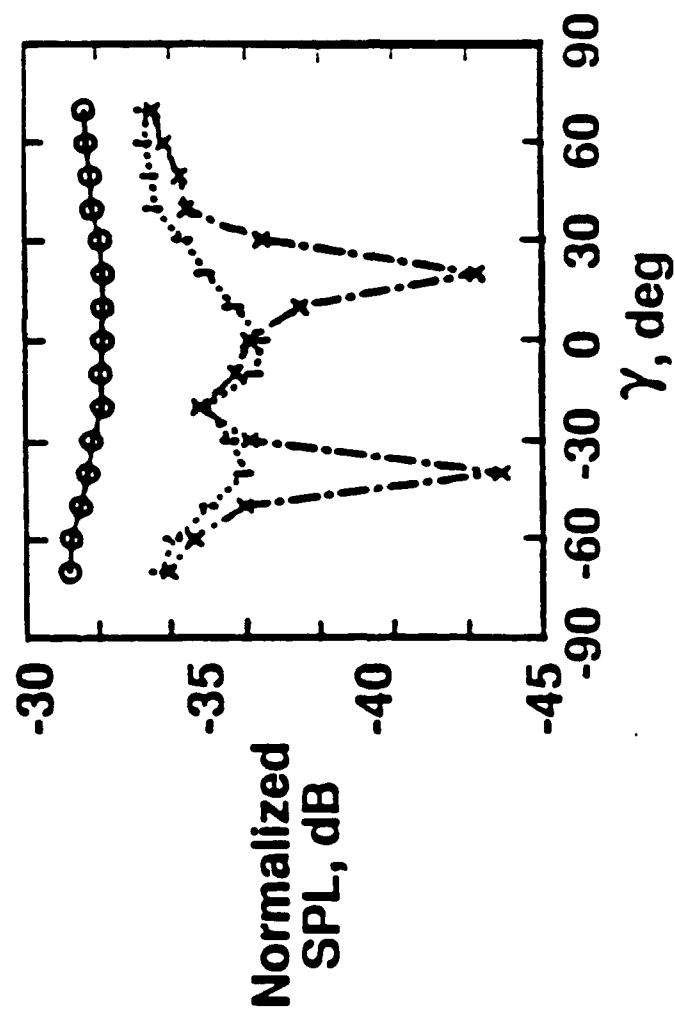


Figure 5. Radiation directivity, microphone error sensors.
 —○—, no control; - - - x - - -, one control; — - - x — - -, two controls.

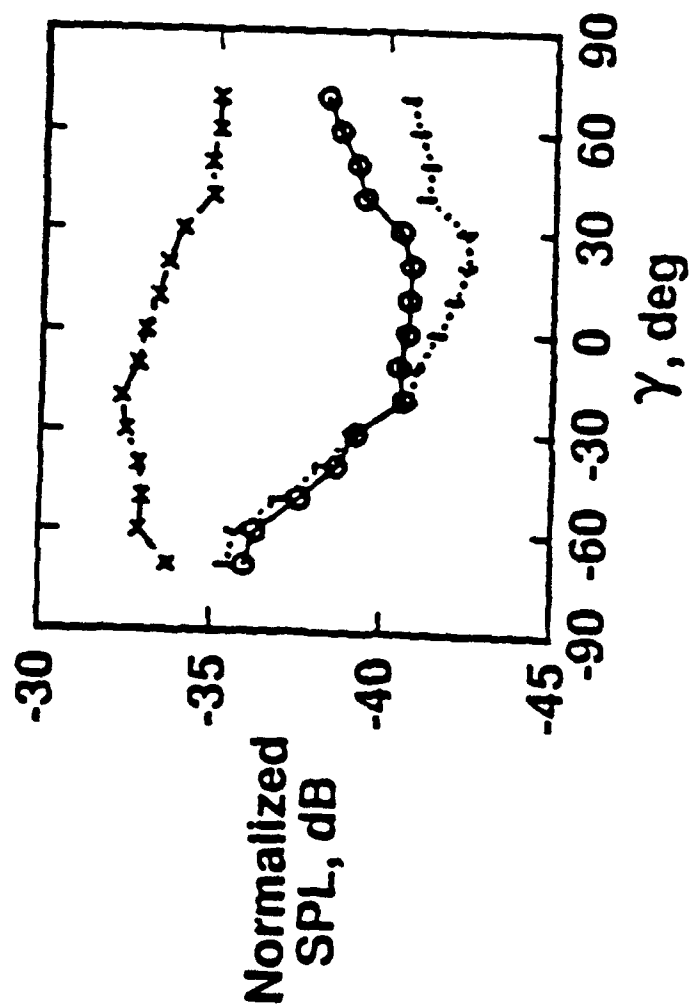


Figure 6. Radiation directivity, accelerometer error sensors.
 —○—, no control; - - -, one control; — — —, two controls.

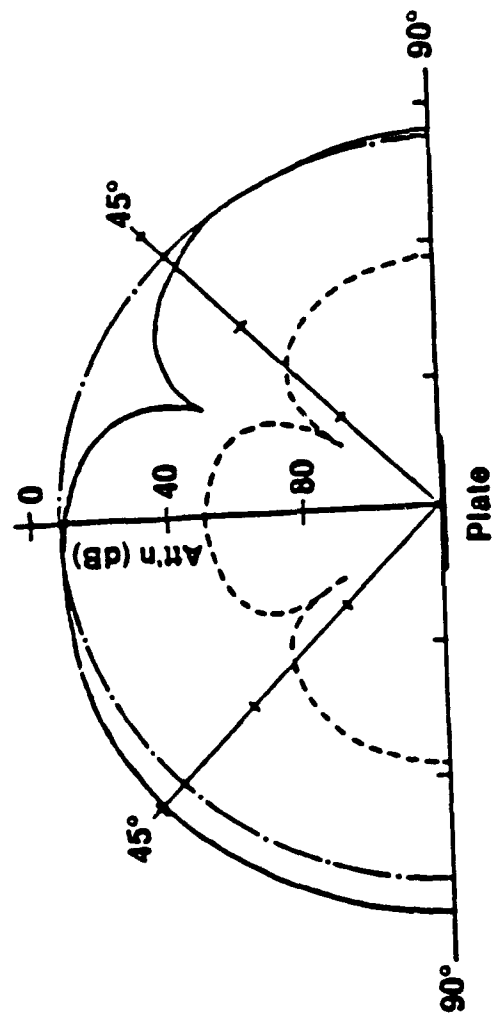


Figure 7. Radiation directivity. —, no control; — · —, one control; - - -, two controls.

function is the total power radiated from the panel rather than minimizing at discrete points [14]. Significant increases in sound attenuation are observed when the number of actuators is increased from one to two while the corresponding panel displacement shown in Figure 8 predicts little change in overall panel amplitude of vibration. Figures 9 and 10 give experimental results for radiation directivity and amplitude of panel modes of vibration for control of sound radiation from a vibrating rectangular panel located in a baffle by a single point force [15]. The error sensor is a microphone situated in the radiated acoustic field. The panel is being driven off-resonance. In this case the panel was manufactured from steel with dimensions of 380 x 300 x 10 mm and was excited at a frequency of 1698 Hz, by a non-contacting electromagnetic exciter. Again the results show significant reduction in radiated sound with relatively little change in panel amplitudes of vibration. It is important to uncover the mechanisms occurring in these cases. The key lies in the results of Figure 10 which show that the relative temporal phases of the uncontrolled modes have significantly changed. Thus the action of the controller has been to alter the relative phases of the modes with the result that the total residual vibration shape is more complex and has a lower radiation efficiency. More exactly, the modes have been adjusted so that their total radiation impedance is lower leading to a fall in radiated sound power [16]. We call the first mechanism where a dominantly radiating mode(s) is reduced in amplitude, control by modal suppression, and the second where the plate response does not reduce but has a lower value of σ , control by modal restructuring [15].

Under this Grant VPI&SU has further refined and expanded the development of active structural acoustic control and addressed a number of the issues raised in the preceding discussions. These developments demonstrate a considerable advancement in the understanding of active structural acoustics and the design procedures for more optimal acoustic designs. The following paragraphs will highlight this work and present appropriate results.

4.1 MULTIPLE ACTUATORS

The number and position of actuators have been varied to determine the effects of control authority using multiple actuators [17]. A test plate, which was mounted with simply supported boundary conditions in a rigid steel frame, was cut from steel and measured 380 x 300 x 1.96 mm, and PZT actuators were positioned for two cases as shown in Figure 11. The first plate configuration allows coupling with odd modes while the actuator positions for the second were optimally selected to couple with the (3,1) mode of the plate (i.e., at the antinodes where surface strain is the highest for this mode). Modal analysis of the plate agreed well with plate theory (i.e., resonant frequencies were within 1%), and the plate mode shapes are illustrated in Figure 12. As in the previous experiments microphones were used as error sensors in the radiated field as shown in Figure 13, and the experiments were conducted inside VPI&SU's anechoic chamber. A signal generator was used to drive a shaker as the disturbance source and to provide the reference signal to the controller. These interconnections with the "filtered-x" version of the Widrow-Hoff adaptive controller are illustrated in Figure 14, but the controller itself is more fully discussed in the control approach part of this report (Section 7.2). Experiments were performed for both on- and off-resonance cases of the plate.

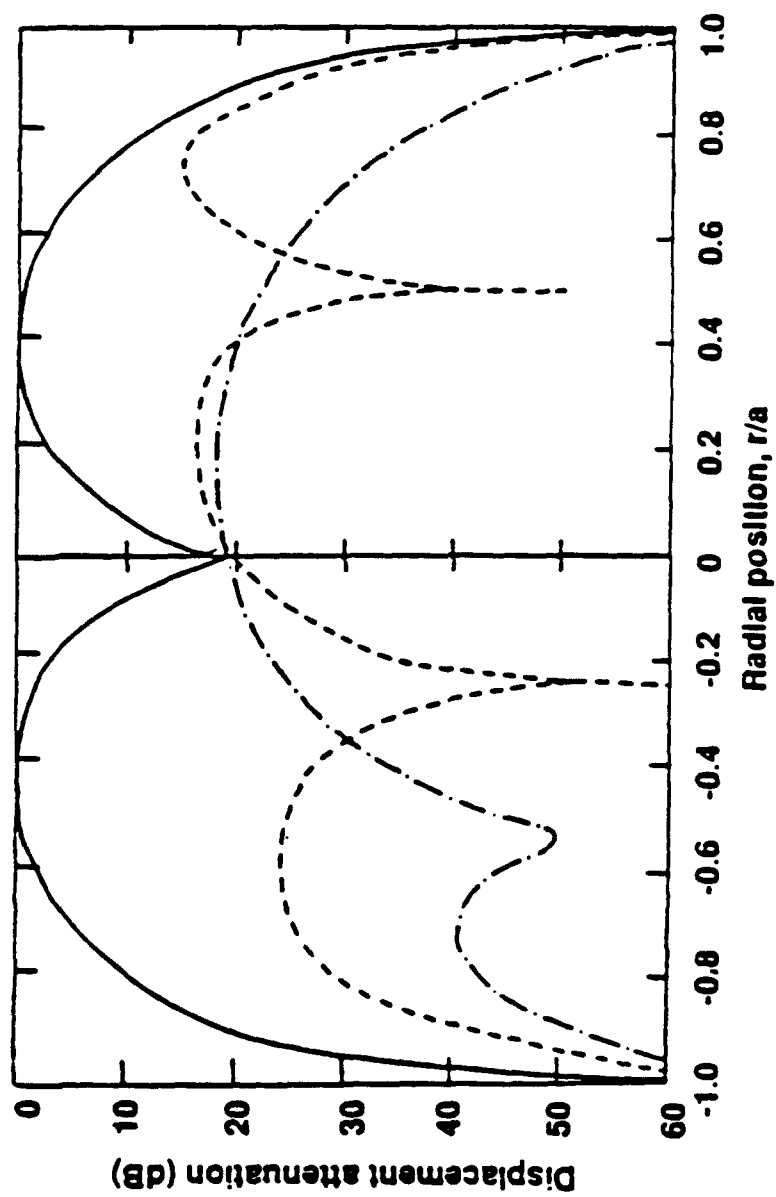


Figure 8. Panel displacement. —, no control; —·—, one control; - - -, two controls.

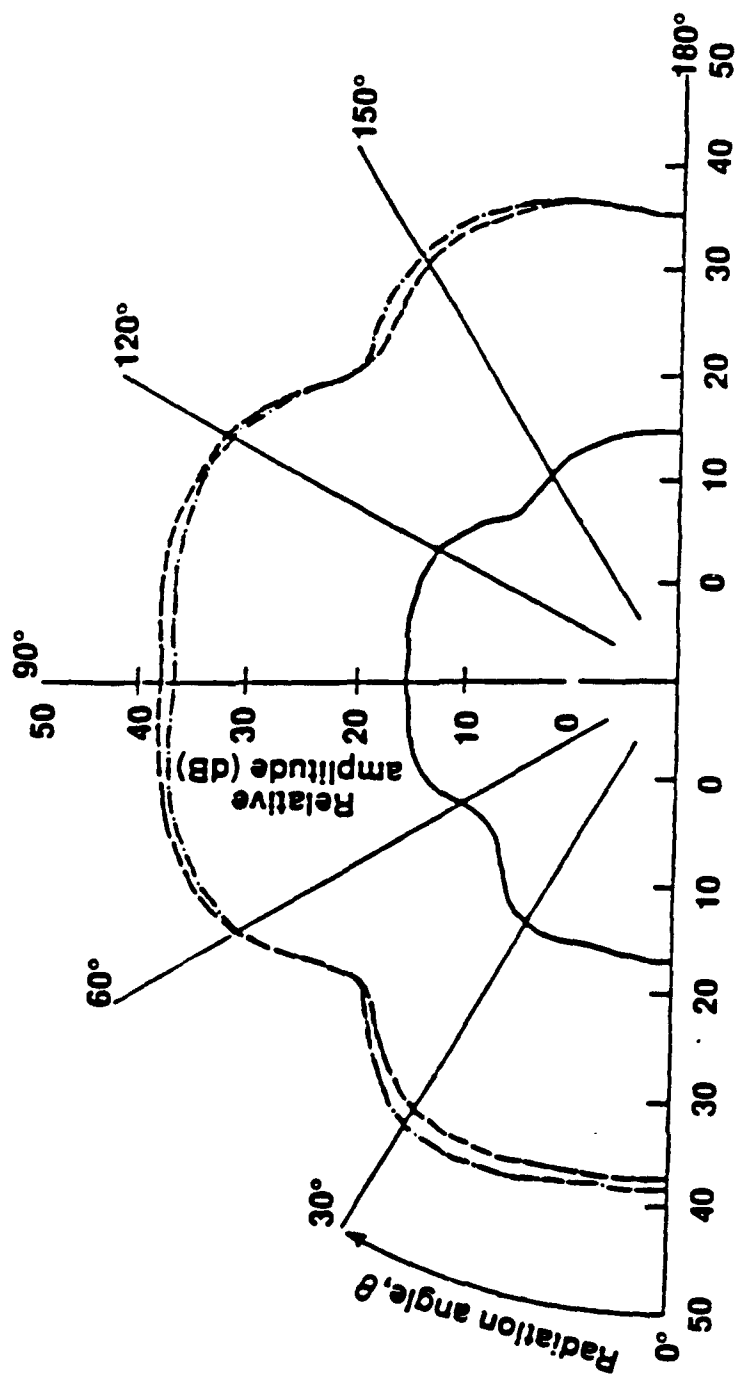


Figure 9. Radiation directivity. —, no control; - - -, control field; —, with control.

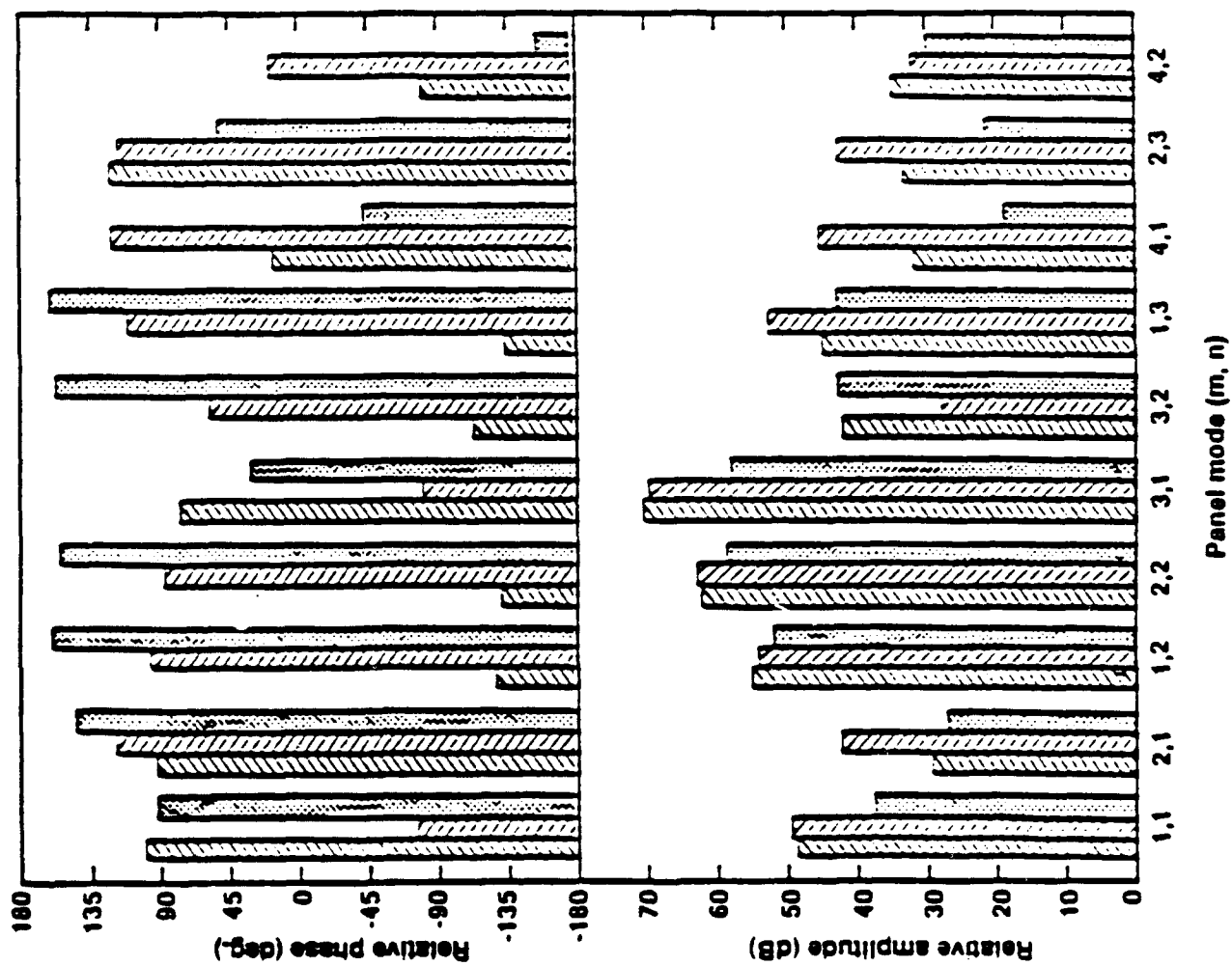


Figure 10. Panel modal response distribution.
 ///// , no control; , control field; x x x x , with control.

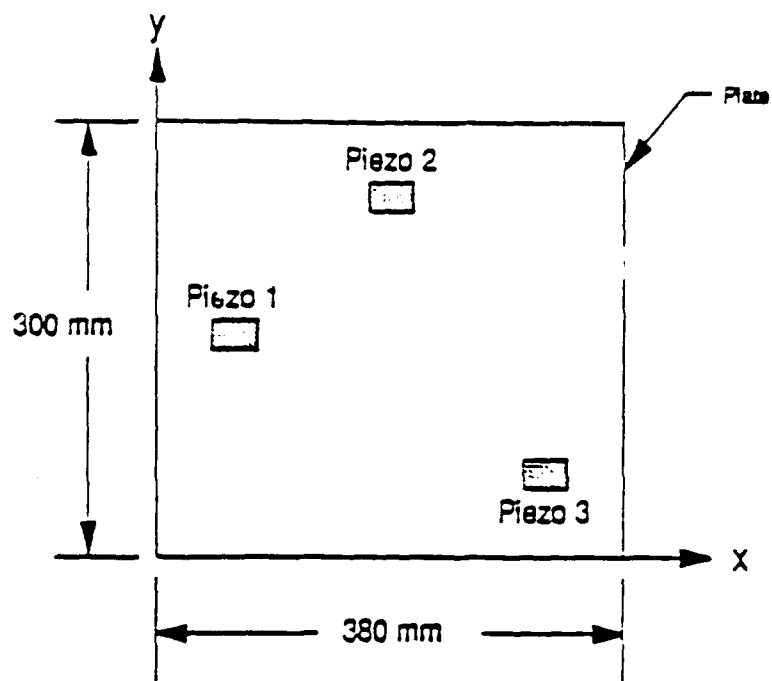


Plate 1

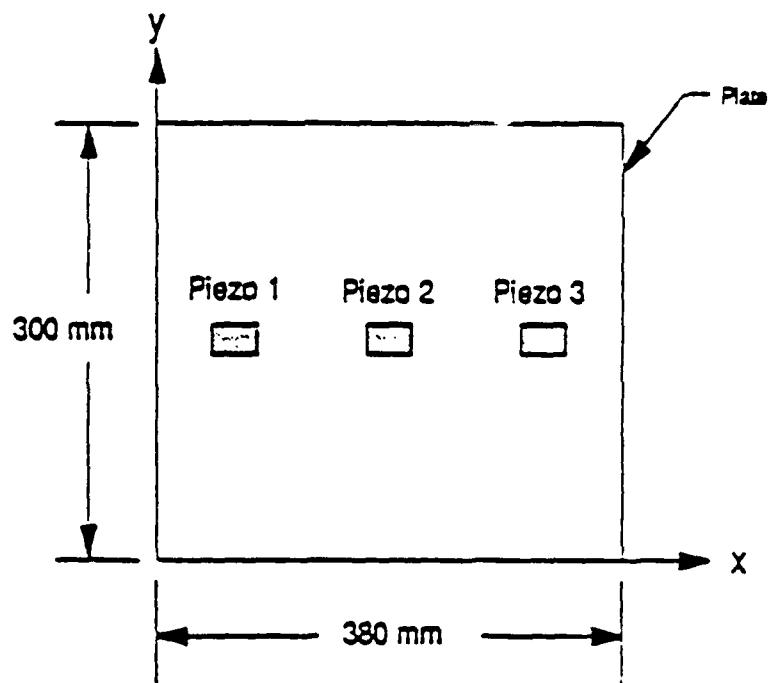


Plate 2

Figure 11. Schematic of Plates 1 and 2.

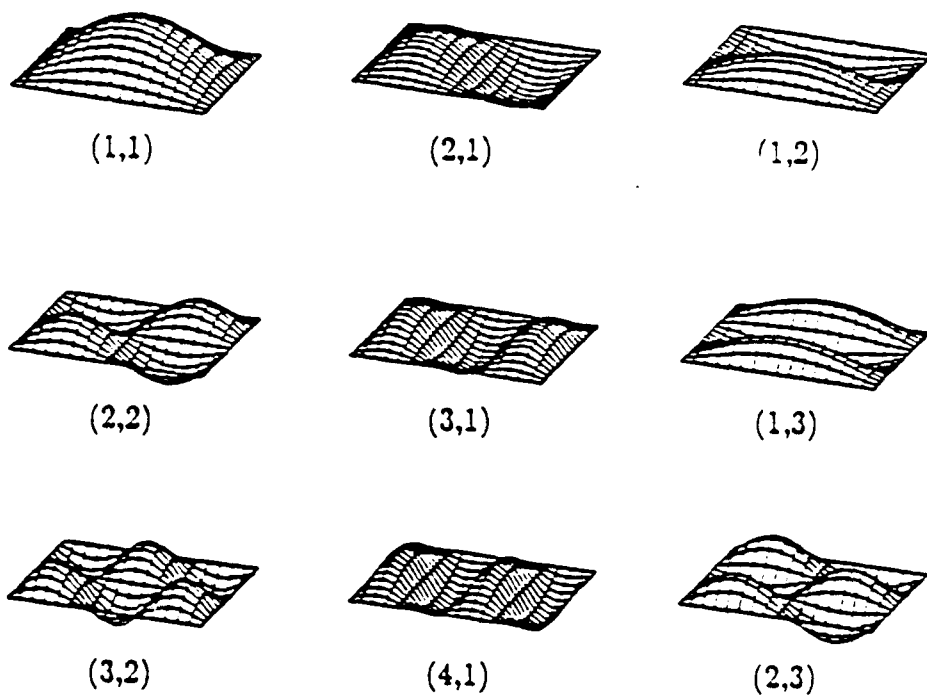


Figure 12. Schematic of mode shapes (m, n) .

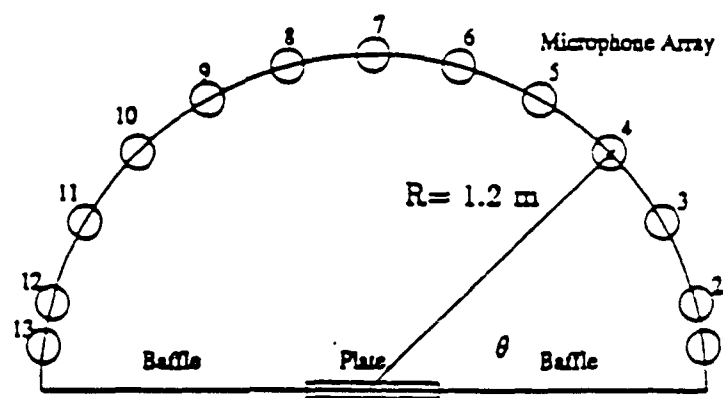


Figure 13. Schematic of microphone array (Top View).

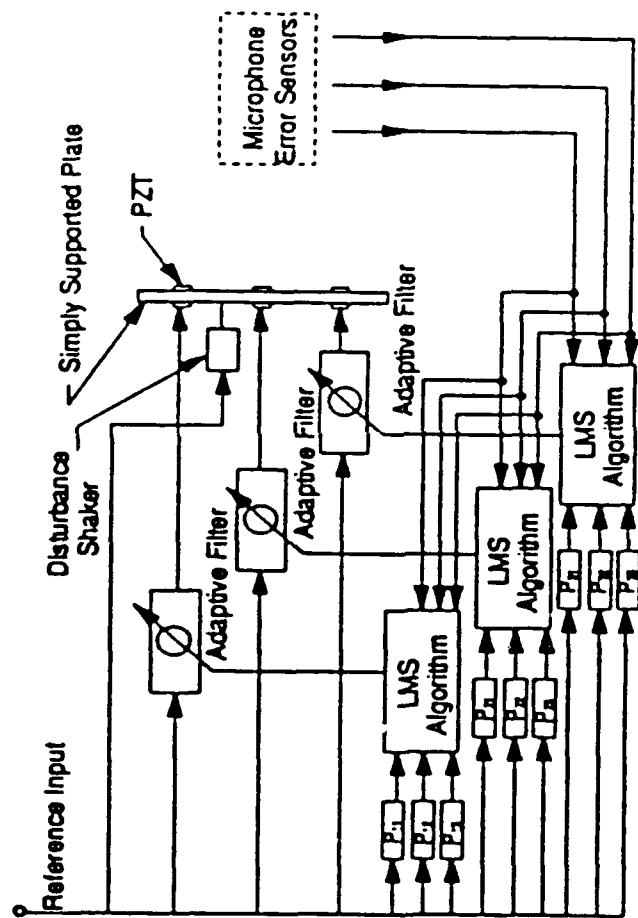


Figure 14. Experiment configuration.

For the first case, the Plate 1 (see Figure 11) was driven on the resonance for the (3,1) mode which is an efficient radiator, and both the acoustic and modal responses are shown in Figure 15. The acoustic far field was attenuated between 30 and 40 dB with only marginal improvement while increasing the number of control inputs ("C#" indicates actuator used, and "M#" indicates error microphone used). With one actuator acoustic attenuation was achieved by modal reduction, but as the number of actuators increased modal restructuring appears. In the second case Plate 1 was driven at 400 Hz, and the results are shown in Figure 16. Here significant improvement is seen with increasing number of control inputs. As the modal amplitudes increased with control attenuation was achieved by modal restructuring (i.e., modes phased for constructive interference in the far field). The second plate (Plate 2) was constructed for the study of on-resonance excitation and control of mode (3,1). However, for off-resonance cases, control authority was observed to be poor since the coupling into odd modes in one direction was practically impossible. Figure 17 presents the results for 400 Hz, and it can be seen that there was significant attenuation at microphone 7 while the global results are poor. In this case control spillover into mode (3,1) was significant, and undesirable since it is an efficient acoustic radiator. This shows the importance of actuator location for the control of acoustic radiation, and optimization approaches have been developed which will be discussed later in this report.

4.2 WAVENUMBER DOMAIN

Wavenumber domain approaches for active control have been investigated analytically and experimentally [18, 19, and 20]. For the plate used in preceding experiment taking the 2-D wavenumber transform of the plate-baffle system one can show that modal suppression corresponds to a general fall in the amplitude of plate response across the complete wavenumber spectrum. However, for modal restructuring there is a reduction in supersonic (propagating) wavenumber components and an increase in subsonic (nonpropagating) components [20] as conceptually illustrated in Figure 18. As discussed in Ref. 1, only supersonic wavenumber components radiate to the far-field, and thus although the averaged panel response is approximately the same or possibly greater, the radiated far-field sound levels fall globally.

Control cost function implementations based upon wavenumber domain objectives have been addressed, and suggest that sensors can be shaped to observe only the supersonic wavenumber components can yield a minimization of the acoustic structural response [20]. Global minimization of total radiated power from a plate for would require a 2-D integral evaluated over the region $-\kappa$ to $+\kappa$. For one of the preceding plate experiments (Plate 1 Figure 11) selecting the radiation in a plane normal to the plate (corresponding to κ_x) for minimization as a test case, the κ domain cost function control simulation results are shown in Figure 19. Wavenumber analysis of structural vibrations has proven to be powerful alternative tool for studying the mechanisms of control [21].

Another interesting observation is that, using a stationary phase approach to evaluate sound radiation from panels, it can be shown that only one spectral wavenumber component of the complete structural motion will lead to radiation towards one particular spatial angle [18]. Thus,

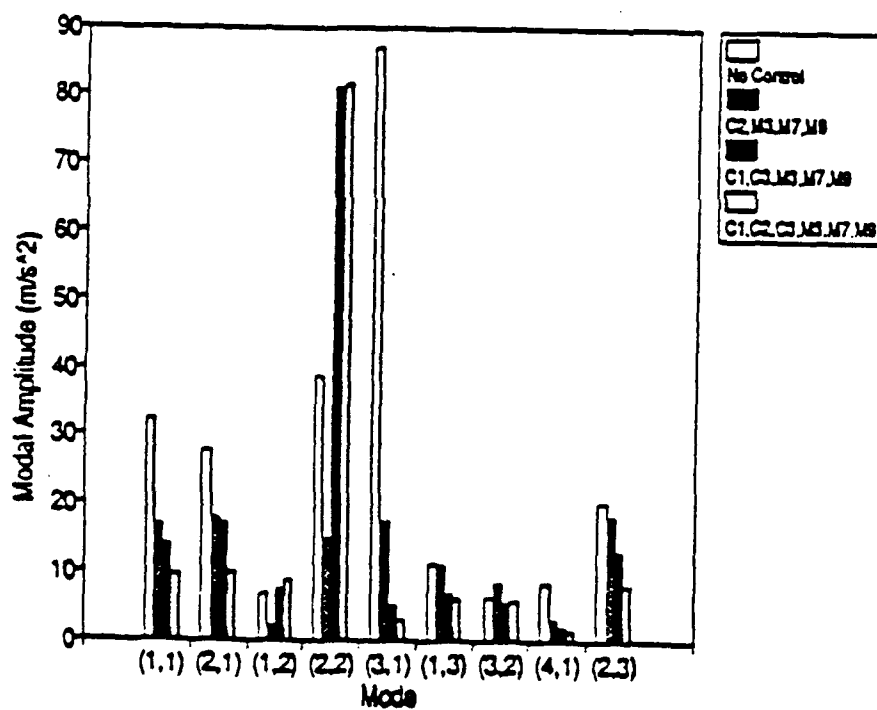
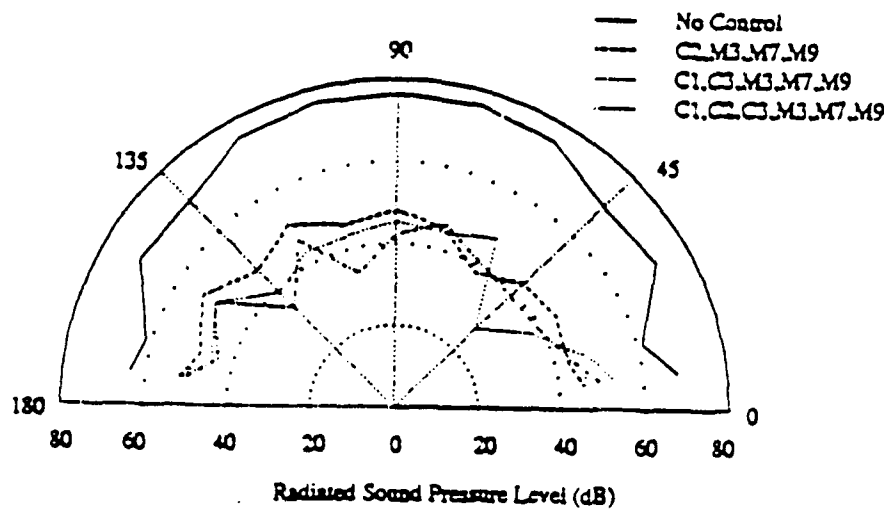


Figure 15. Response at (3, 1) mode (Plate 1).

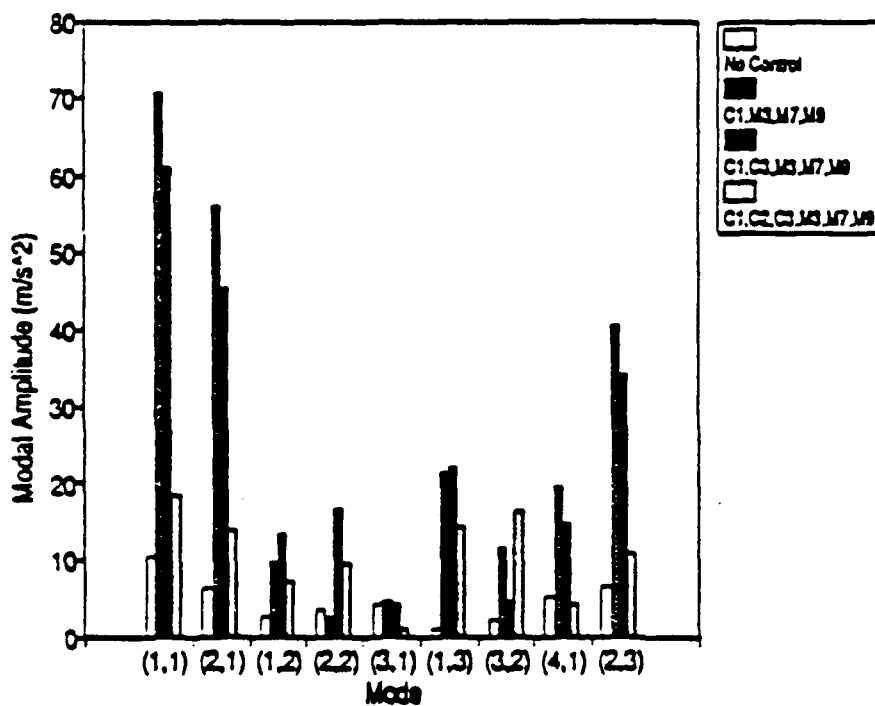
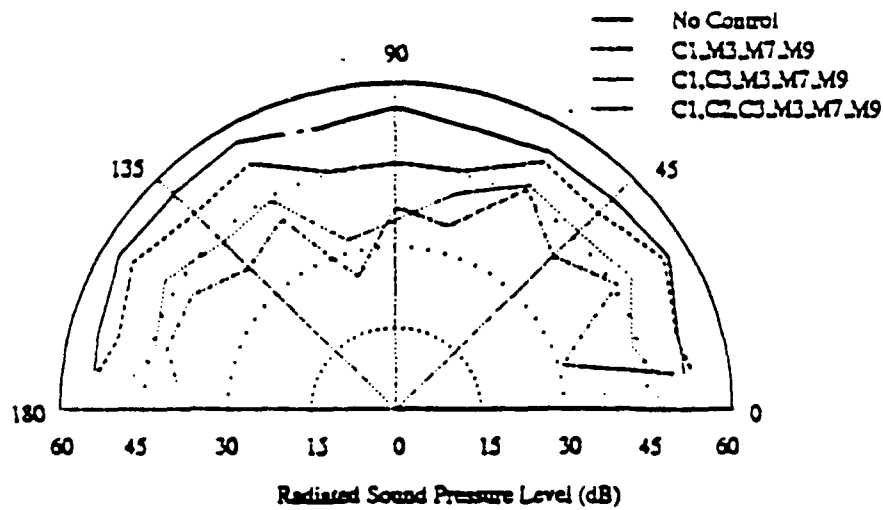


Figure 16. Response at 400 Hz (Plate 1).

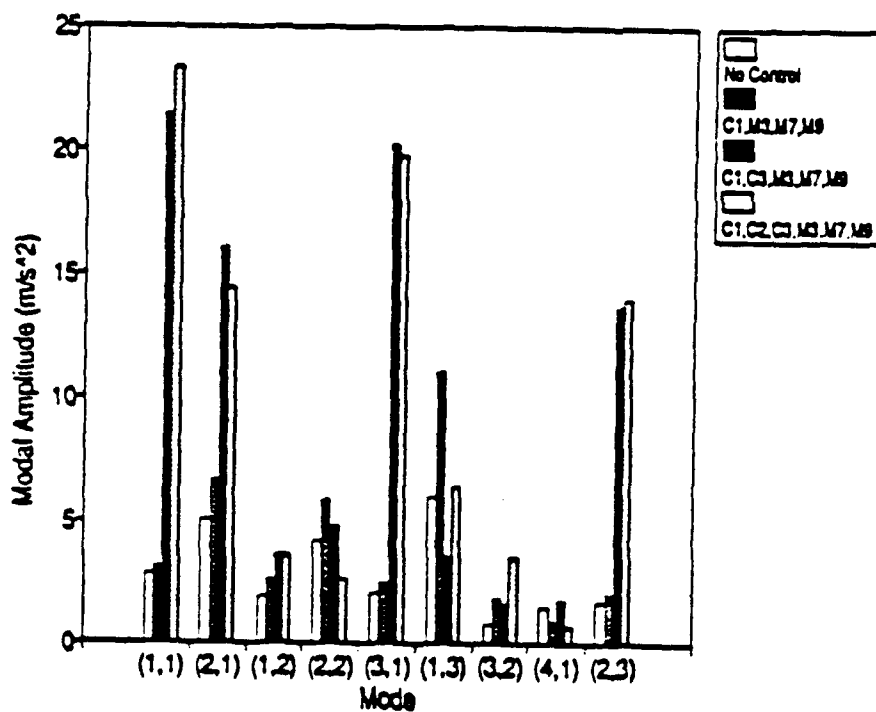
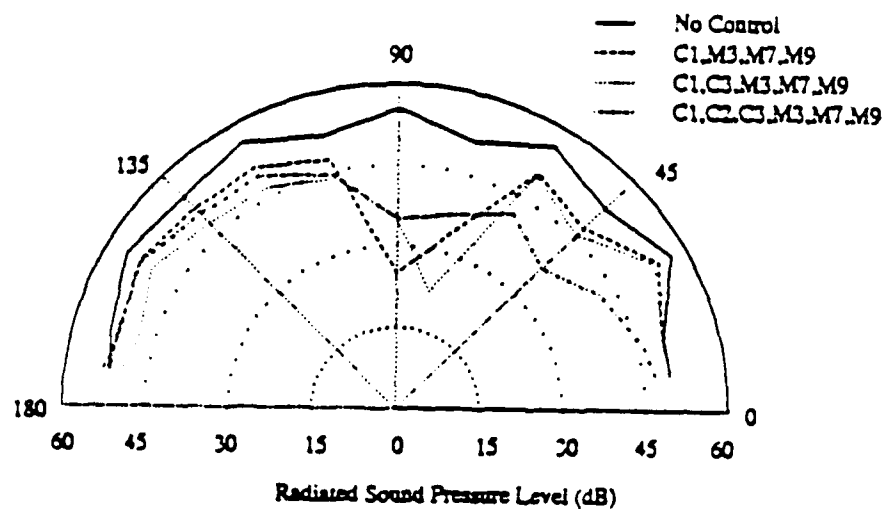


Figure 17. Response at 400 Hz (Plate 2).

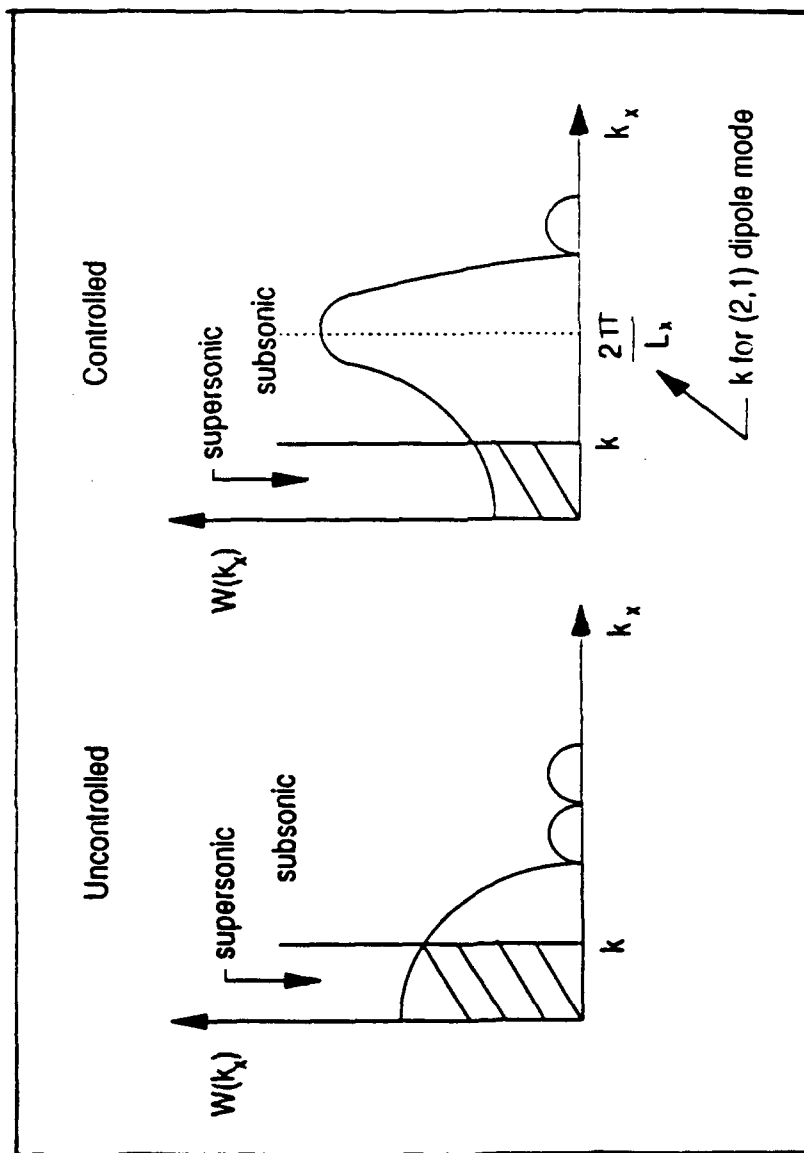


Figure 18. Example of 1-D κ - transform.

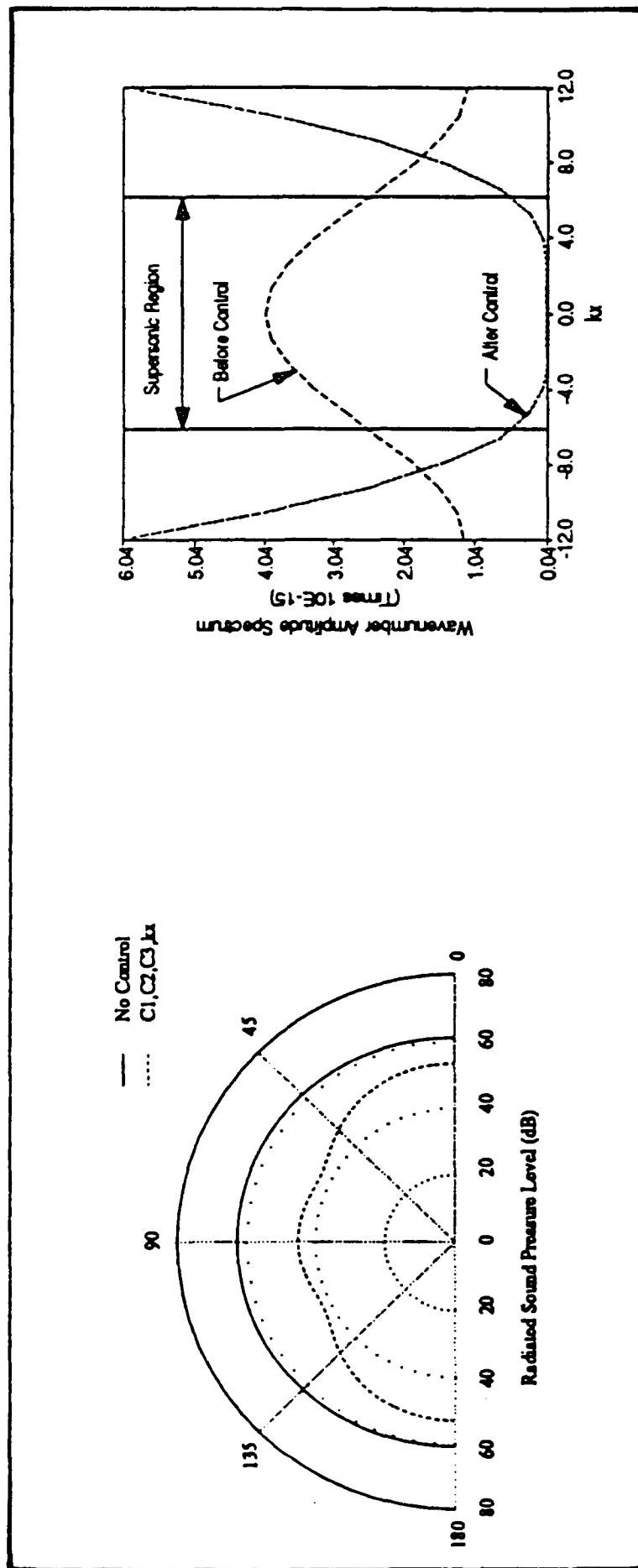


Figure 19. Response at 320 Hz (κ - transform Cost Function).

if one designs an optimal controller to minimize particular isolated wavenumber components on the structure, sound radiation can be minimized at the corresponding radiation angle., without the use of an error microphone in the radiation far field. Figure 20 shows for example, the radiation directivity pattern results for a 2-D baffled simply supported thin beam of length 0.38 m, bending stiffness 137.879 NM, and mass per unit width 0.238 Ns^2/m^2 driven at the resonant frequency of 285 Hz. In this case, the objective is to minimize radiation at an angle of 45° , and indeed the results demonstrate that the sound is driven to zero at an angle of 45° .

Near-field structural acoustic error sensing (i.e., acoustic sensors on or near the structure's surface) has been investigated as a means of controlling the far-field [22]. This has included the analysis of the near-field pressure distributions, the time averaged radiated intensity distributions under the same conditions to provide insight into the control sensing problem.

4.3 OPTIMAL ACTUATOR/SENSOR PLACEMENT

An algorithm has been developed and evaluated for choosing the optimal actuator/sensor configuration for controlling sound radiation from a baffled simply supported plate [23]. It was applied to optimize the location of a rectangular piezoelectric actuator and both the size and location of a rectangular surface strain error sensor constructed from PVDF. The resulting acoustic responses predicted from analytical models have been compared to experimental results obtained from appropriate measurements in the VPI&SU anechoic chamber. These single optimally located actuator results have been further compared to those from control with a non-optimally positioned actuator as well as multiple control actuators. In addition either microphones are used to provide error information in the test cases or a single optimally located and dimensioned PVDF error sensor is implemented as the cost function.

This algorithm was theoretically obtained as previously outlined by Wang, Burdisso and Fuller [24] using an acoustic objective function consisting of a finite number of microphones approximating the total radiated acoustic power. A flow chart of the solution strategy is presented in Figure 21. The core of the algorithm for determining the optimal piezoelectric actuator location or the optimal PVDF error sensor size and location are identical. Linear Quadratic Optimal Control Theory (LQOCT) is used to compute the optimal control solution for the given actuator/error sensor location in step 1, regardless of whether path 1 or 2 is taken. An IMSL subroutine NOONF, which solves a general nonlinear programming problem, was implemented to compute the optimal solution for both path 1 and 2 of Figure 21. The optimal design parameters for both the actuator and sensor were constrained to meet physical limitations such as plate boundaries as well as limitations on control voltage to the actuator and the availability of specific sizes of the material.

Both the analytical and experimental cases addressed were for 550 Hz excitation of a simply supported plate constructed from steel measuring 380 x 300 x 1.96 mm (used in previously described experiments) and configured with four actuators and three PVDF sensors as shown in Figure 22. The optimum actuator and sensor locations, are appropriately labeled in Figure 22, were based upon the analytical results for the off-resonance 550 Hz excitation. The other

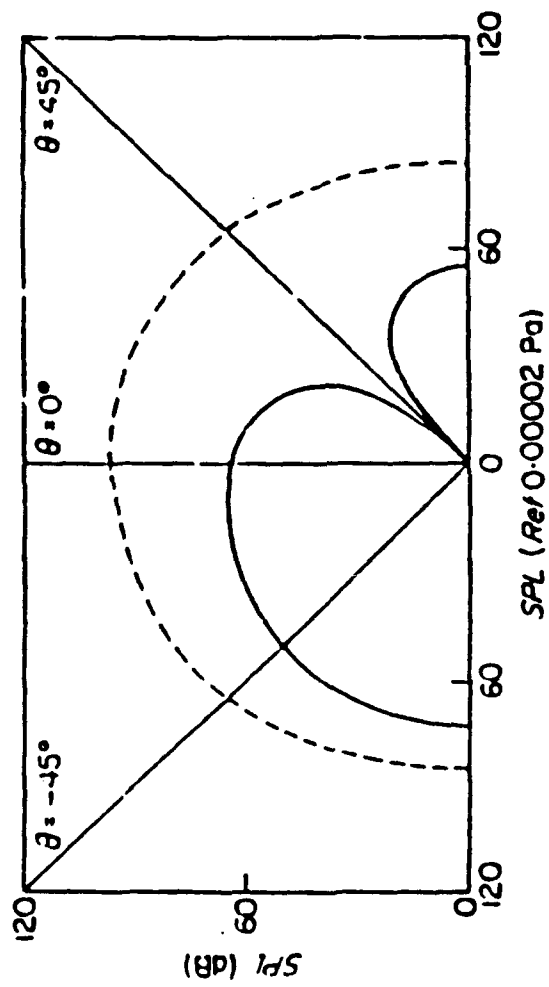


Figure 20. Directivity of radiated sound pressure at resonance (285 Hz).
Minimization of radiated pressure field at $\theta = 45^\circ$:
——, uncontrolled; ———, controlled.

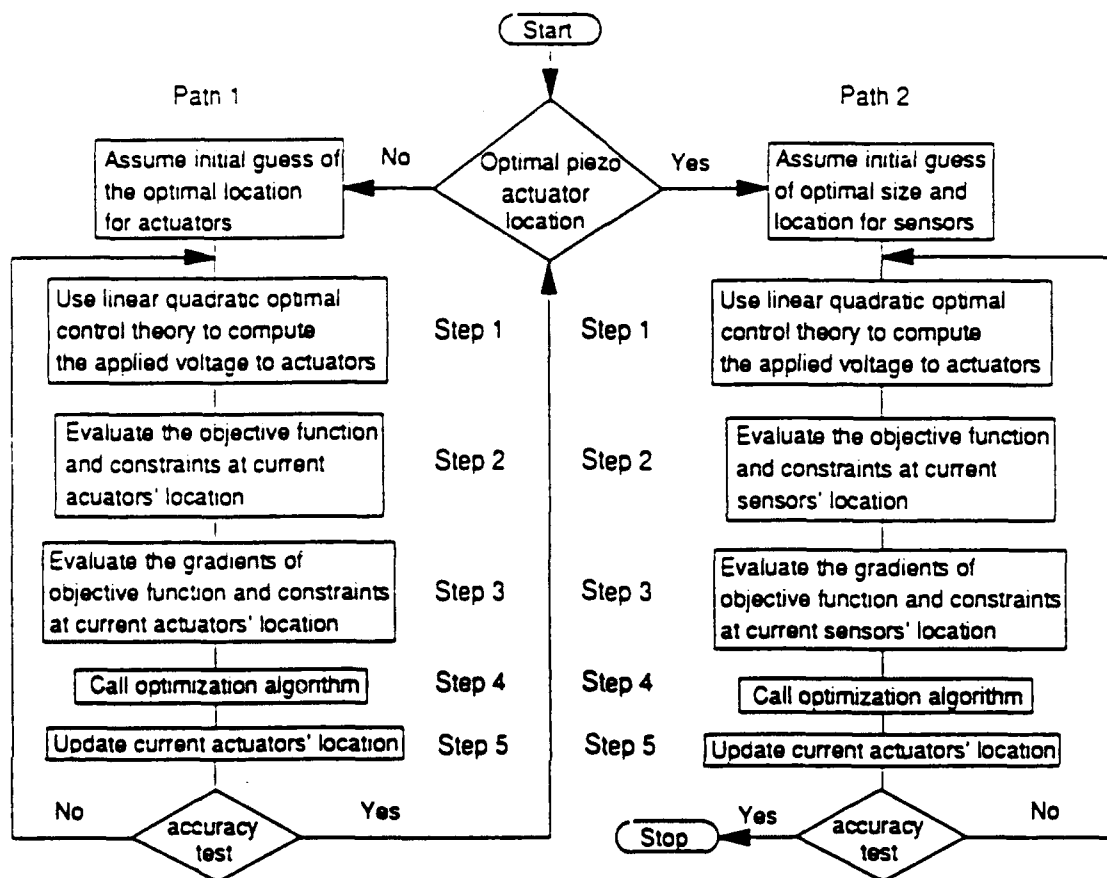


Figure 21. Flow chart of optimization algorithm.

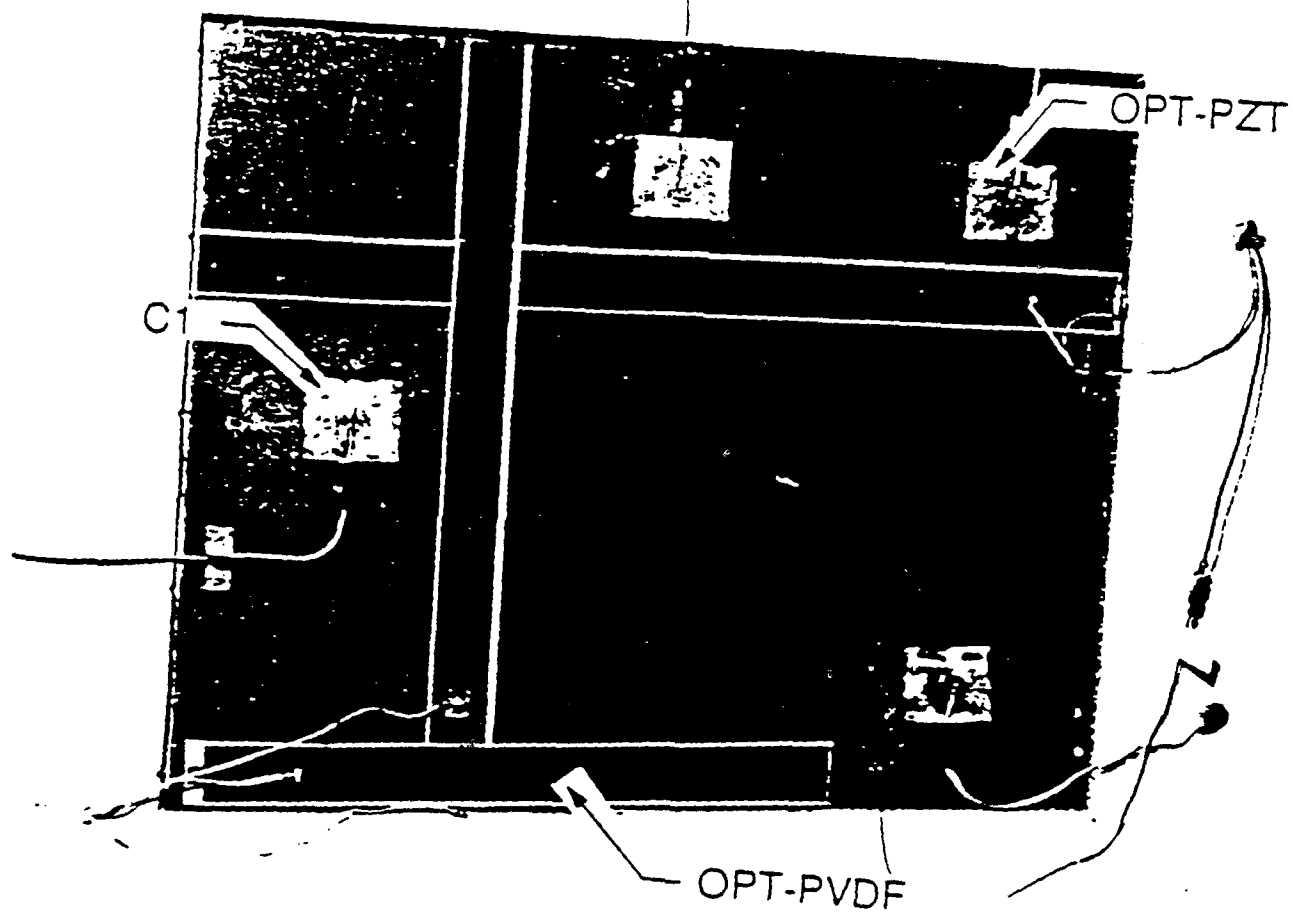


Figure 22. Picture of test plate with actuators and sensors.

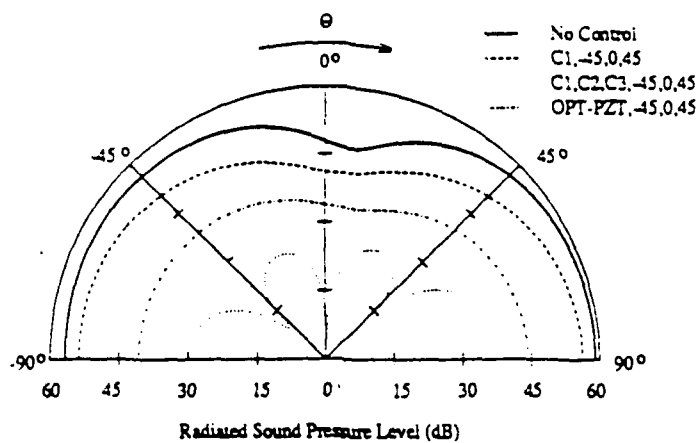
actuator and sensor locations used in previous studies are for comparison purposes. The disturbance was generated using a shaker on the back of the plate at spatial coordinates (230, 130) mm.

The analytical and experimental results presented in Figure 23 demonstrate the level of acoustic control which can be achieved with a single optimally located piezoelectric actuator. Error sensors used in the test case were three microphones at a radius of 1.6 m, and at angles designated for convenience 45° , -45° , and 0° in the figures displayed. In the first case, a single, non-optimally located actuator (denoted C1) was used to achieve control. In the second case, all three non-optimally located actuators (denoted C1, C2, C3) were used for control, and finally the optimally located actuator (denoted OPT-PZT) was used. The single optimal actuator location performs about 10 dB better than the non-optimal locations. The experimental results confirm the predicted results with even more attenuation observed along the axis near the baffle which should not fully be considered since the baffle was truly finite.

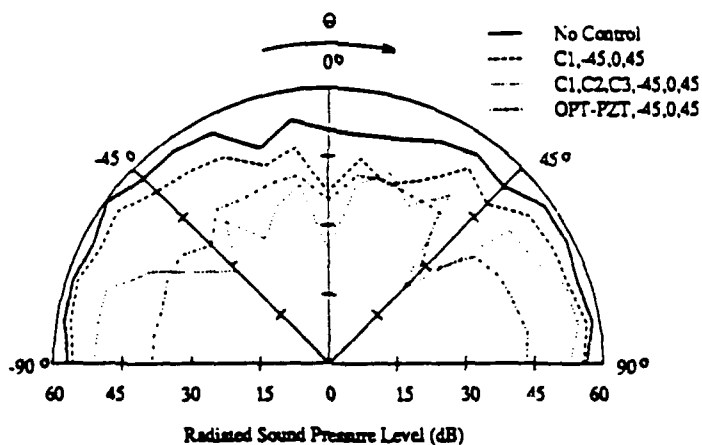
The analytical and experimental results are shown in Figure 24 for using the optimal PVDF error sensor rather than the three microphones used previously. In the first case, a non-optimal located actuator (denoted C1) is used with the optimally oriented PVDF sensor (denoted OPT-PVDF). The results are about 5 dB, and comparable to those using of microphone error sensors in Figure 23. When the optimal actuator location and sensor locations are used (denoted OPT-PZT, OPT-PVDF) the performance increases to about 15 dB. This clearly demonstrates the significance of the optimization. In the final test case of Figure 24, the acoustic response when using microphones with the optimally located actuator (denoted OPT-PZT, $45,0,-45$) is compared to that from the optimal actuator/sensor pair the predicted and measured acoustic responses yield similar trends. In terms of total acoustic power the control implementation with the optimal actuator/sensor configuration resulted in attenuation within 0.1 dB of that predicted when the optimally located actuator is used in conjunction with microphone error sensors.

4.4 NASHUA NUMERICAL MODEL

Active structural acoustic control has been demonstrated in the literature for several elementary structures using analytical models. However, analytical approaches cannot be used for the practical and important case of a 3-dimensional structure immersed in a dense fluid, which occurs primarily in marine applications. Such fully coupled problems, in which appreciable fluid-structure interaction takes place, require a numerical approach. Submerged shells are of significant practical interest, and for a general, 3 dimensional fluid loaded shell no such analytical expressions exist. The research presented here uses a numerical approach to develop a general algorithm for investigating active structural acoustic control of submerged structures [25]. Predictions of the dynamic response and radiated noise field were obtained using the computer program NASHUA developed by Everstine [26]. VPI&SU is cooperating with David Taylor Research Center (DTRC) Bethesda, MD to extent the capabilities and applications of the NASHUA model. NASHUA uses the finite-element program NASTRAN to compute structural quantities, and a boundary-element formulation to solve for the fully coupled structural acoustic response. There are two significant advantages to using an approach based on NASHUA. First,

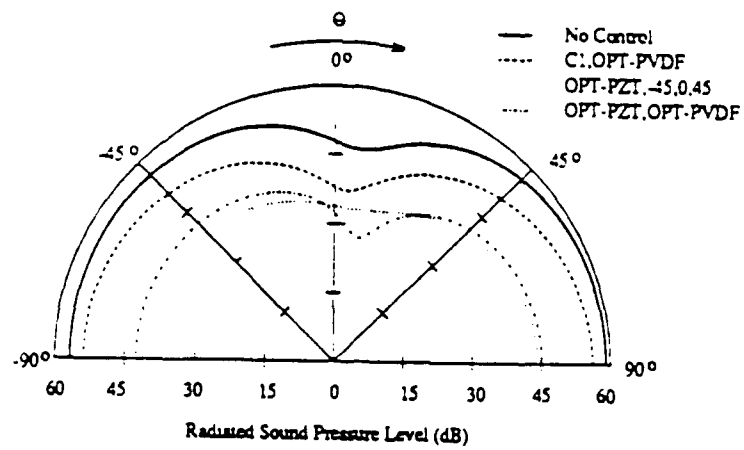


(a) Predicted acoustic directivity pattern

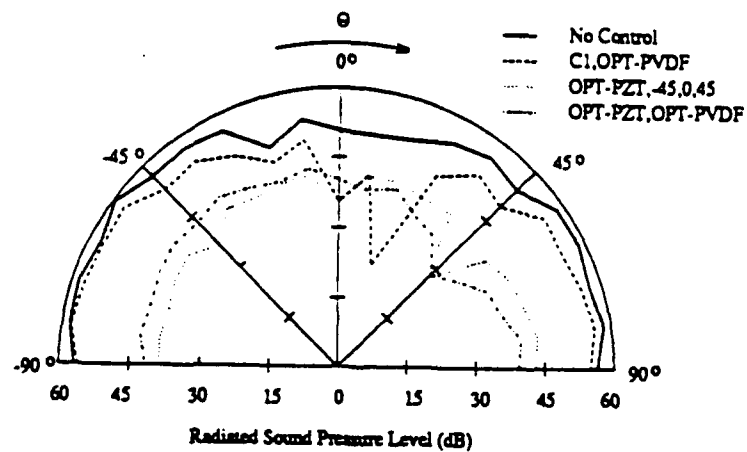


(b) Measured acoustic directivity pattern

Figure 23. Comparison of theory and experiment for optimization of actuator location.



(a) Predicted acoustic directivity pattern



(b) Measured acoustic directivity pattern

Figure 24. Comparison of theory and experiment for optimization of actuator and sensor location.

no discretization of the ambient fluid is required since the fluid is modelled using boundary elements. Second, the approach can be used for any structure that can be modeled using NASHUA.

The control algorithm used is a feedforward method in which require *a priori* knowledge of the nature of the disturbance. After specifying the number and locations of the control actuators, Linear Quadratic Optimal Control Theory (LQOCT) is used to solve for the complex optimal actuator forces that minimize a quadratic cost function. Two separate cost functions have been evaluated and compared. The first is a spatial domain cost function that approximates the radiated power. This is appealing and guarantees global farfield pressure reductions, but requires error sensors in the acoustic farfield and would therefore be difficult to implement in some cases. The second cost function is a wavenumber domain (κ - domain) controller similar in concept to that discussed by Fuller and Burdisso [18]. This cost function requires no farfield information, although computing the wavenumber spectrum requires some extra computer time.

The method is illustrated using a thin, fluid loaded spherical shell for which there are analytical solutions for certain excitation types. The results presented here only address steady-state, single frequency forcing functions, and to further simplify the analysis only forces applied in a global plane are considered. This simple case serves as a benchmark with which to develop methodologies and computer programs in anticipation of more complex structures. The closed form solution also serves to validate the numerical results. The finite element portion of the NASHUA model uses axisymmetric plate elements that model both bending and membrane stresses. The nonuniform structural mesh contains 129 grid circles with smaller spacing near the two poles. The poles are not truly closed, but approximated by grid circles of very small diameter. This implies that the point force is approximated by a normal ring force applied at the pole grid circle. For the normal point force excitation, there exists an analytical solution for the dynamic responses of a thin spherical shell as presented by Junger and Feit [1]. Figure 25a shows the result for a single force as a function of frequency, and the levels agree well throughout the frequency range. In Figure 25b, the radiated power is broken into its modal components, which can be found using the analytical solution.

The simplest control setup is to place a single control force at ($\Theta=180^\circ$), directly opposite the noise force ($\Theta=0^\circ$). By optimizing the complex control force at each frequency, the radiated power spectra shown in Figure 26a is obtained. The radiated-power and wavenumber cost functions are denoted Π and Ψ , respectively. Figure 26a shows the radiated power vs. frequency with each of the controllers, and also without control. Figure 26b is similar except that the quantity is the wavenumber cost function. Figure 27 shows the optimum control forces for the two controller cases. In both cases, a single control force reduces the cost function 10 dB or more at resonant frequencies, and the reductions between resonances are much smaller.

Controller performance can depend strongly on both the number of actuators and their locations, and the control algorithm developed above can easily handle multiple control forces. In addition, to examining the effects of actuator locations, the approach can be used to optimize the actuator locations as well as the control forces vs. frequency. For the simple cases presented

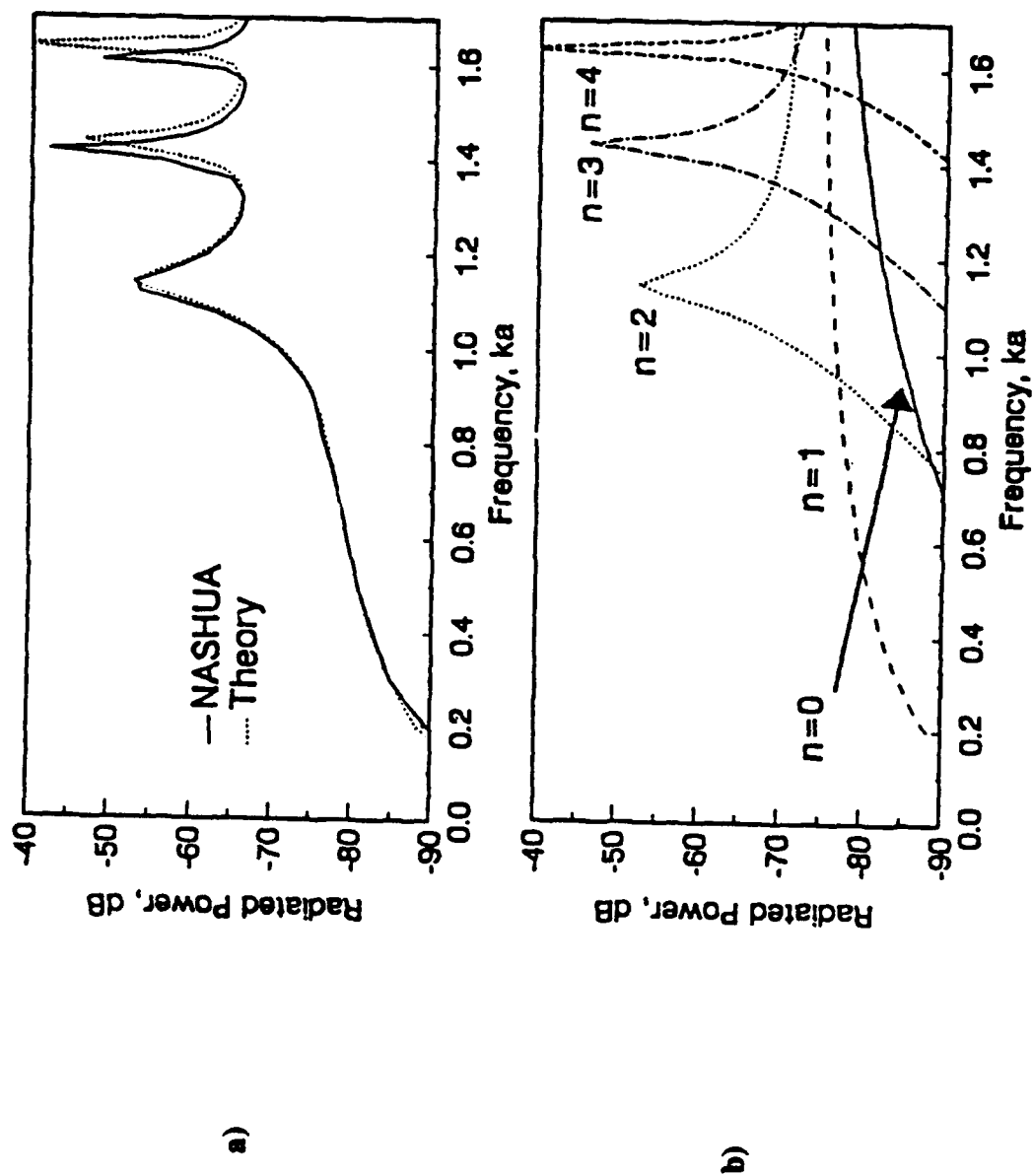


Figure 25. Response to point-force excitation.
a) Radiated power vs. frequency.
b) Modal components of radiated power vs. frequency.

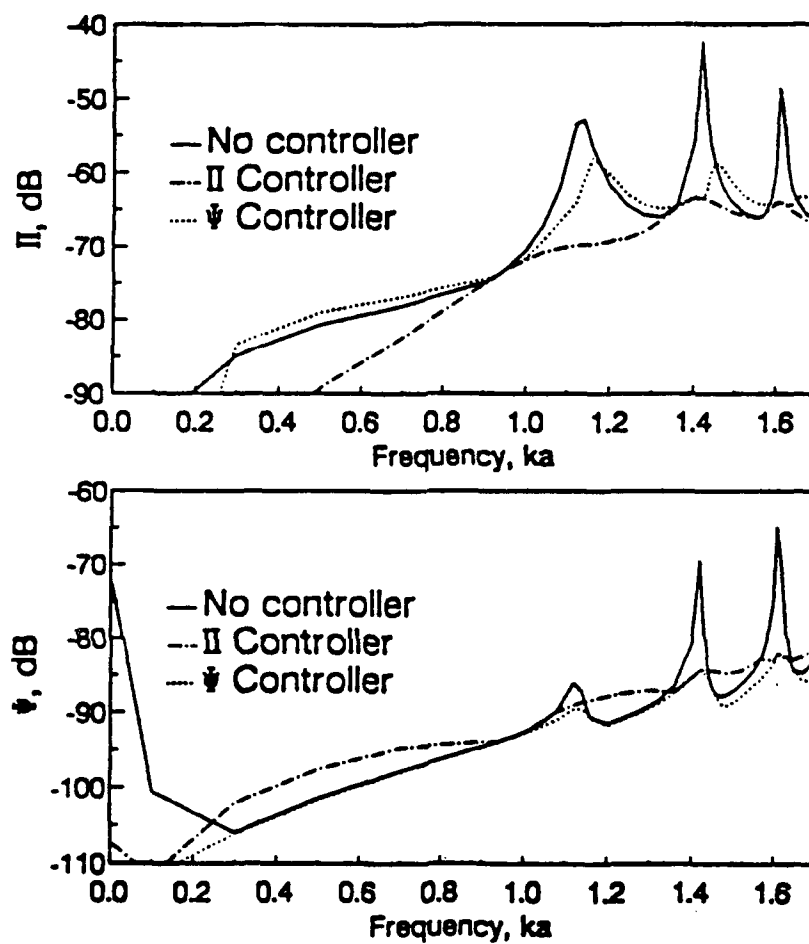


Figure 26. Controller performance with one control force.
a) Π vs. frequency. b) Ψ vs. frequency.

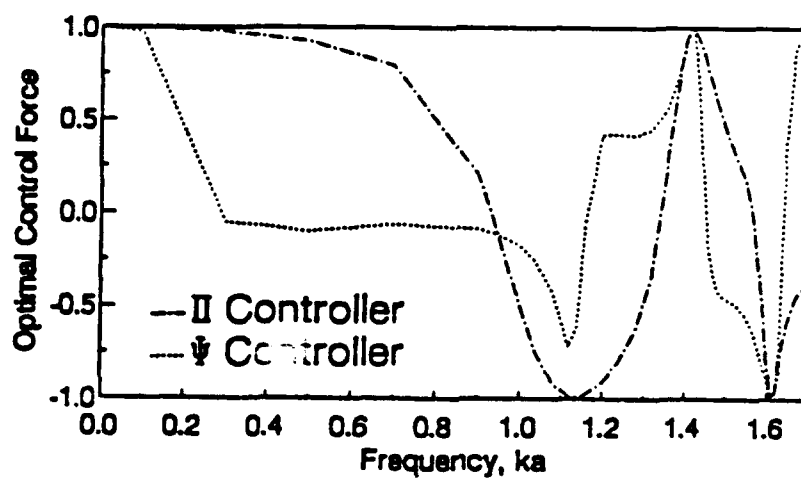


Figure 27. Optimal control force vs. frequency.

here the computation times are so short a number of cases can be examined by trial and error for the best locations. To illustrate this a pair of point forces that resemble a concentrated moment (i.e., a force of +1.0 at $\theta=0^\circ$ combined with a force of -1.0 at $\theta=5^\circ$) is considered. Figure 28 shows the radiated power vs. frequency without control and with three different control scenarios. The single control force at 180° is barely visible because it lies on top of the no controller case. The variable location point force results are for optimum actuator locations determined by trial and error at each frequency.

4.5 STRUCTURAL DISCONTINUITIES

Active control of sound radiation due to subsonic wave scattering from discontinuities represented by a line constraint or by a uniform reinforcing rib positioned on a fluid loaded infinite plate has been analytically studied in the spectral κ domain [27]. This considered the use of feedforward control with secondary line forces applied near the plate discontinuity, and the optimal solution of a quadratic cost function based upon the total acoustic power into the fluid half space. The plate was assumed to be loaded with fluid on one side, and "in vacuo" on the other. The first discontinuity is a line constraint at which the displacement is constrained to zero, and the second discontinuity consists of a reinforced rib that acts as a lumped mass with both translational and rotational inertia attached to the plate. These configurations are illustrated in Figure 29. Example active control results for the line constraint and ribbed cases are shown in Figures 30 and 31 respectively. In these cases there are two control forces being applied near the discontinuity, and the disturbance is an incident flexural subsonic wave at a frequency equivalent to values of $\omega/\omega_c = 0.2$ (ω_c is the critical system frequency). These results demonstrate that for both discontinuity cases, significant reductions in the radiated sound pressures can be achieved with two control forces located near the discontinuity. The radiation reduction is strongly dependent upon frequency and location of the control forces, and influenced by the type of discontinuity. Most importantly the acoustic radiation can be significantly reduced using structural control forces. This work has been extended to address finite plates and other boundary conditions where similar good performance of the ASAC technique has been analytically demonstrated [28].

4.6 NRL EXPERIMENTS

As part of the cooperative research with NRL, experimental hardware was provided by VPI&SU to conduct a water tank experiment complete with a actuated aluminum plate with three PZT actuators, sensors, and controller [29]. These experiments investigated the active control of structural acoustic properties with fluid on one side of the plate and air on the other. The results obtained excellent narrowband global control on- and off-resonance for this situation with heavy fluid loading. Control of supersonic plate components were observed using the NRL holographic processing facility.

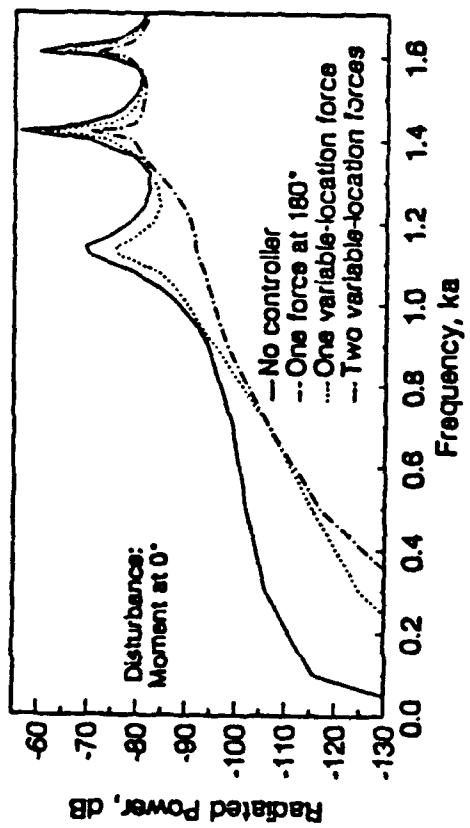
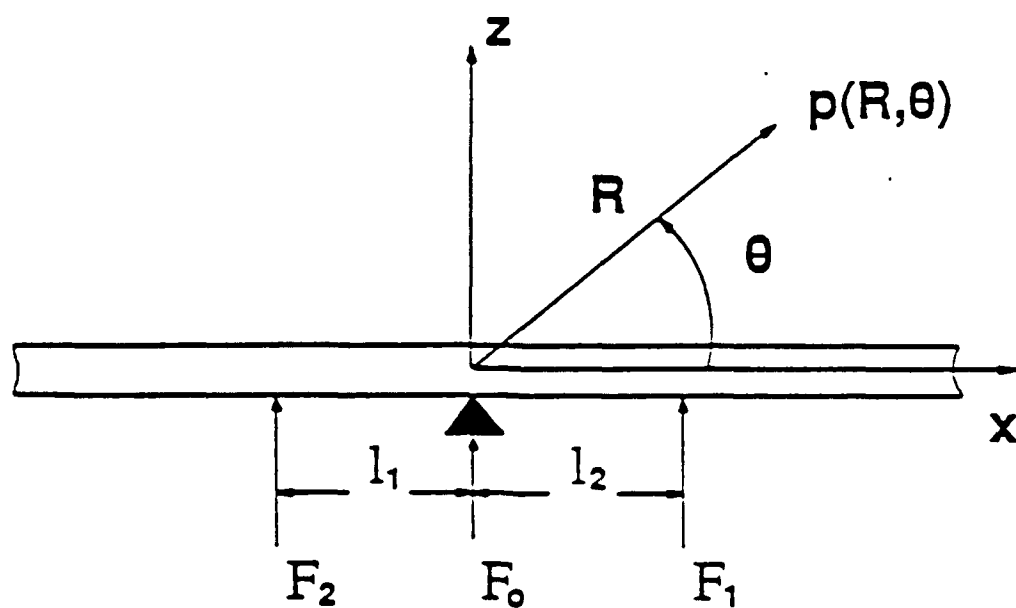
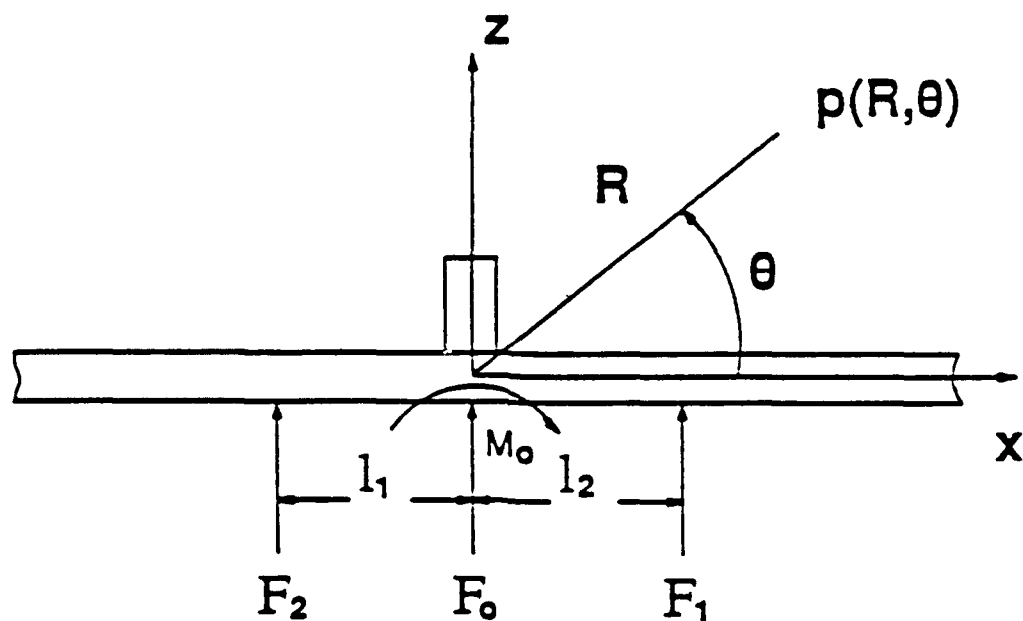


Figure 28. Control of a concentrated moment using fixed-position and variable-position actuators.



(a) Line Constraint Discontinuity



(b) Rib Constraint Discontinuity

Figure 29. Cross-sectional view of structural discontinuities.

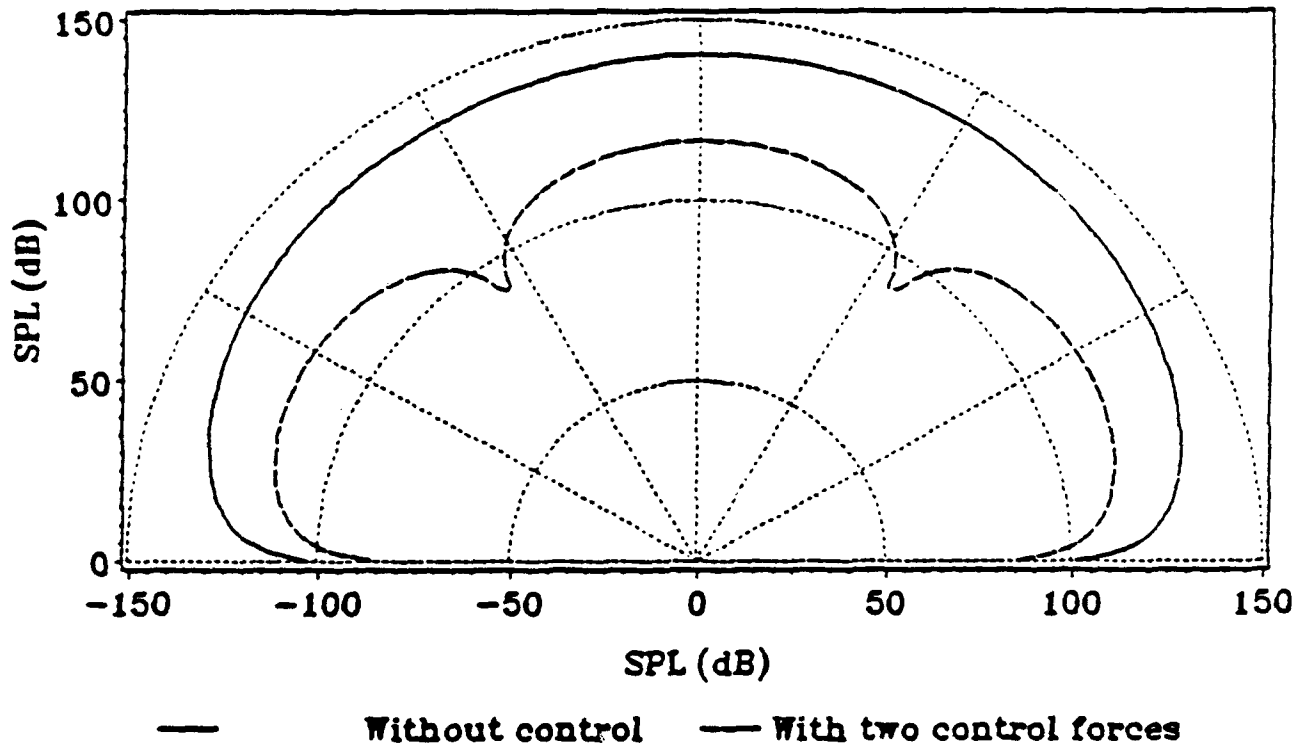


Figure 30. Directivity pattern of radiation - the line restrain case: two control forces.
 $\omega/\omega_s = 0.2$, $l_{1,2} = \pm 0.02$ m, $R = 10$ m, $w_0 = 1$ mm.

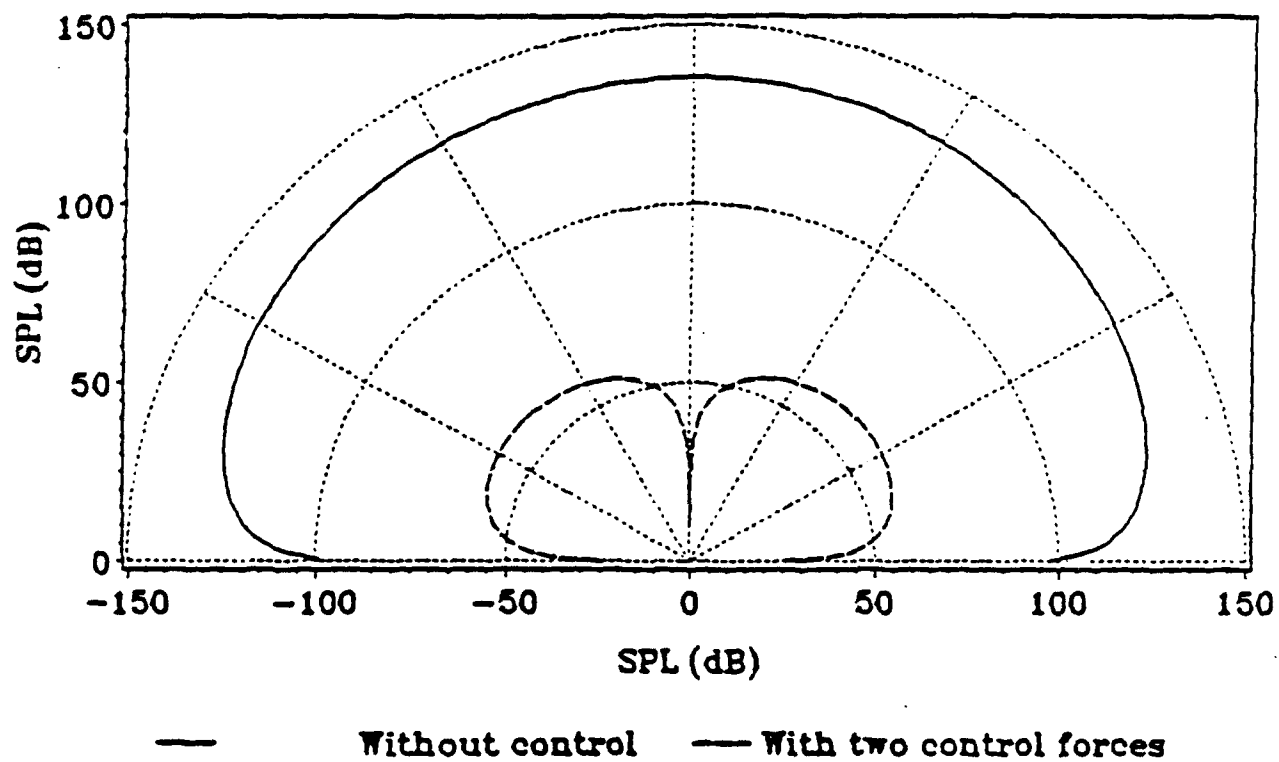


Figure 31. Directivity pattern of radiation - rib case: two control forces.
 $\omega/\omega_s = 0.2$, $l_1 = \pm 0.0001$ m, $l_2 = -0.0001$ m, $R = 10$ m, $w_o = 1$ mm.

4.7 REVERBERANT SOUND TRANSMISSION

The preceding discussions have focused on the control of radiation from structures that have been excited by disturbance vibrational forces applied directly to the radiating structures. Active attenuation of sound transmission through structures has been addressed as well, and most importantly is the application of structural control forces as the sole means of control authority, and further the application of structural sensing. The following discussions will highlight a number of the accomplishments in this work.

Experiments have been performed to demonstrate active control of sound transmission by an elastic plate in the reverberant field [30]. These experiments for sound transmission by a clamped rectangular plate were performed in the VPI&SU Sound Transmission Loss Facility as illustrated in Figure 32. The plate was excited on one side by a speaker positioned arbitrarily in the source chamber. The speaker is driven at 348 Hz which is below the Shroeder cut-off frequencies of the reverberation chambers (the reverberant fields are not diffuse), but at a resonant frequency of the plate. Two types of distributed polyvinylidene fluoride (PVDF) error sensors were used, non-shaped and shaped, and the piezoelectric actuators were located to couple into the desired plate modes to be controlled as shown in Figure 33. For the PVDF plain sensor 2 in the x direction, only odd-odd and odd-even modes were sensed, and for the PVDF sensor 1 in the y direction, only even-odd and odd-odd modes were sensed. The shaped PVDF nth mode sensor was shaped according to the second derivative of the nth mode shape of a clamped-clamped beam (note: shaped PVDF sensors are more fully discussed later in the sensor section of this report). A control law based upon the filtered-x version of the adaptive least mean squares (LMS) algorithm was used and configured as shown in Figure 34. Microphone sensors in the acoustic far field were used as error sensors for comparison purposes, and to measure the sound pressure level (SPL) to quantify the overall results. The table in Figure 35 shows the transmission reduction in the receive chamber at measurement microphone location 1 (rm1) in dB units. The results show that reductions of 20 to 30 dB can be achieved with the error microphones, depending upon the location of the error microphones (em#) and the PZT actuators used. The microphone location influence on performance is because the field is not diffuse, and the actuator locations are dependent upon the coupling to the resonant mode (3,1) at 358 Hz. The PVDF sensors offer effective transmission attenuation when compared to the microphone results. The shaped PVDF mode sensors are not more efficient than non-shaped PVDF sensors since the plate is excited at resonance, but they should be more efficient at off-resonance sound transmission.

This brief discussion highlights the nature of the structural acoustic coupling in the control approach, and some of the VPI&SU contributions to the field. It is apparent that the controlled behavior of these systems can be viewed in a number of ways. These insights as well as many other aspects of natural sound radiation can be taken advantage of in optimizing such control approaches. The following sections of this report will expand on these points in more detail in addressing actuators, sensors, and controller design approaches, implementations, and experiments.

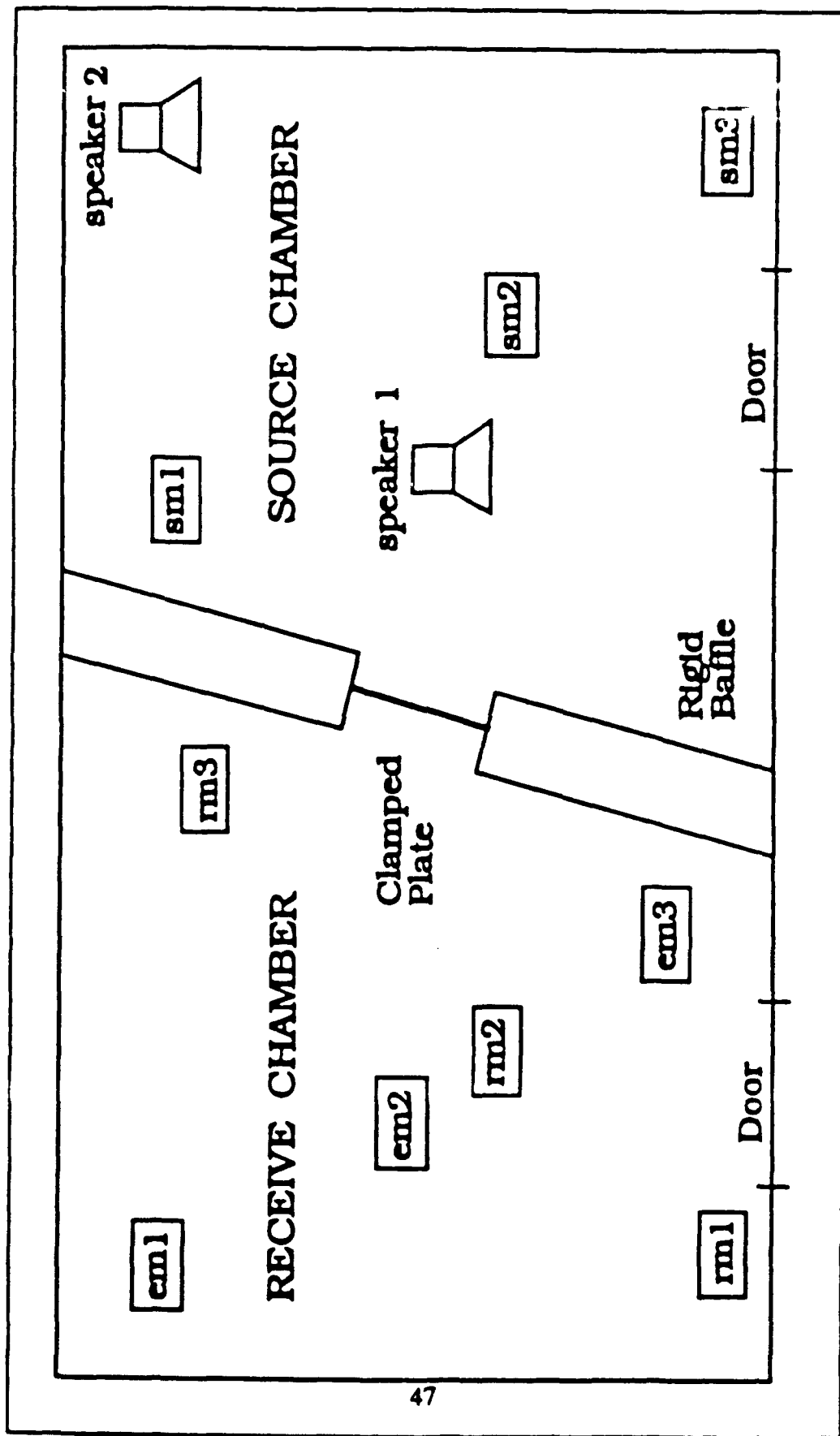


Figure 32. Schematic of reverberation chambers and microphone locations.

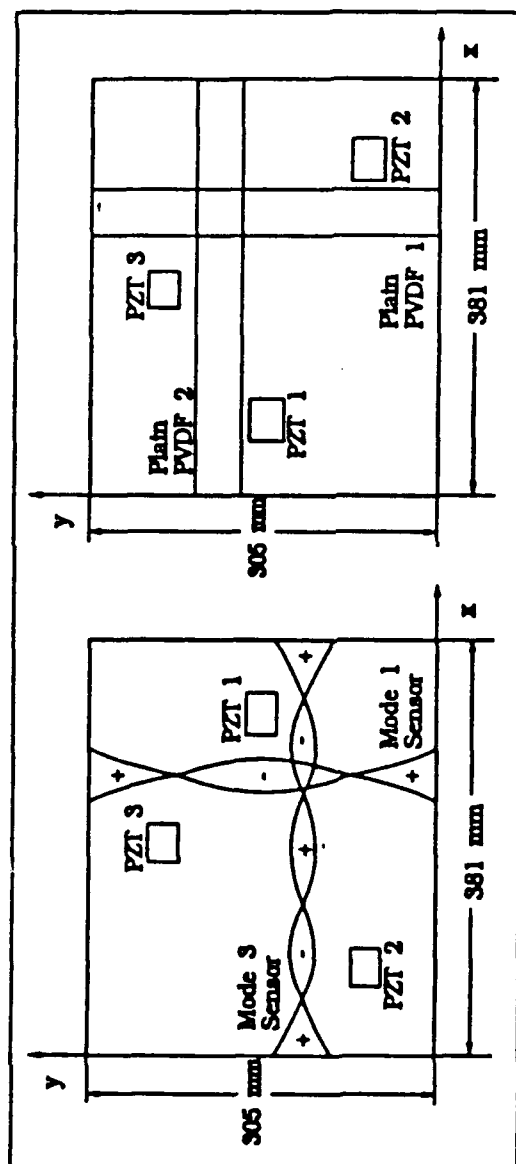


Figure 33. Schematic of test plate.

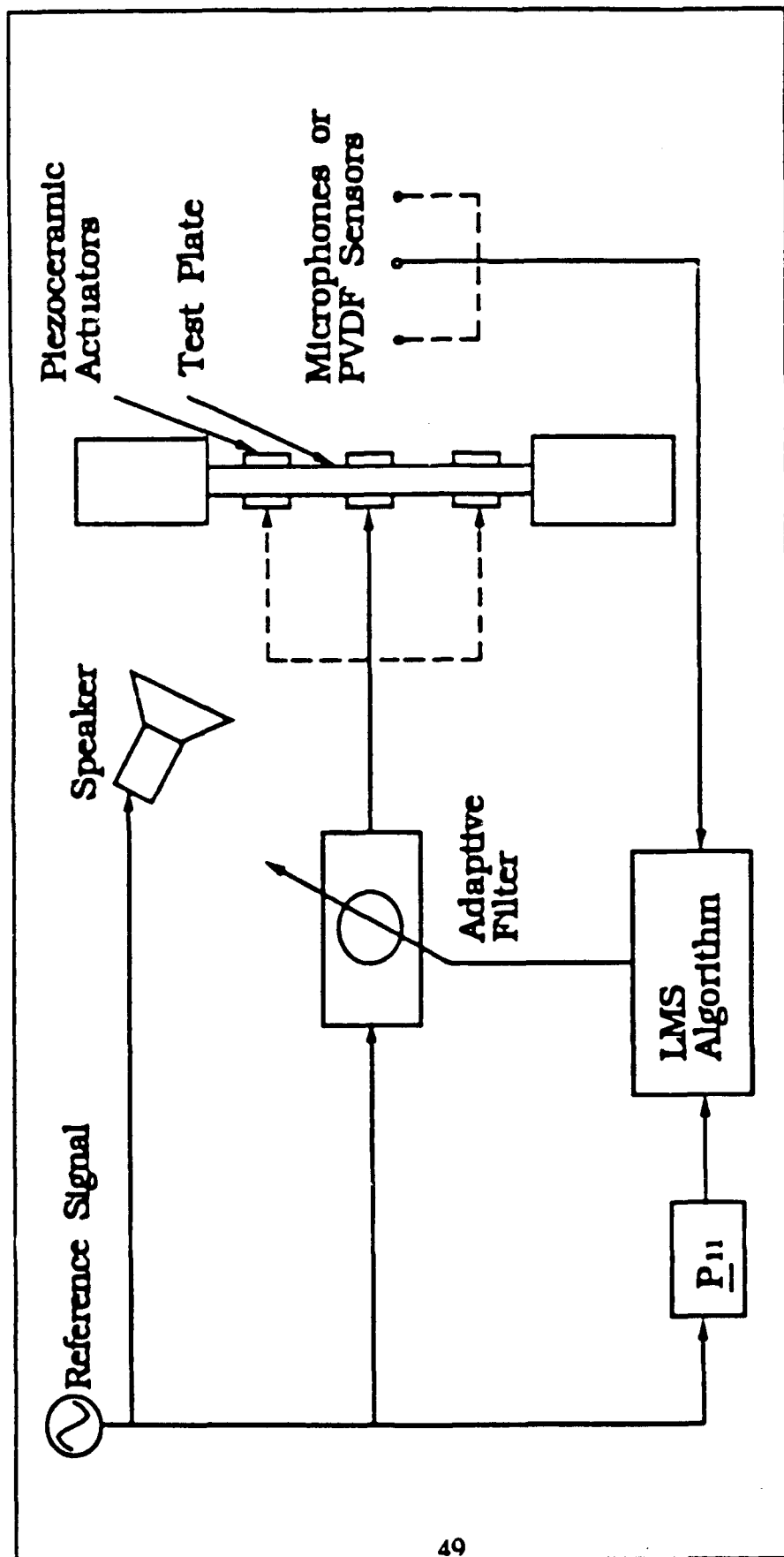


Figure 34. Block diagram of control system.

SOUND TRANSMISSION REDUCTION IN RECEIVE CHAMBER
MEASUREMENT MICROPHONE LOCATION 1 (m1), UNIT: dB

Error Sensors	PZT 1 as Actuator			PZT 2 as Actuator			PZT 3 as Actuator		
	Before Control	After Control	Reduction	Before Control	After Control	Reduction	Before Control	After Control	Reduction
Error mic. 1	92	70	22	92	71	21	92	70	22
Error mic. 2	92	60	32	92	66	26	92	62	30
Error mic. 3	92	67	25	92	72	20	92	66	26
Plain PVDF	92	68	24	92	73	19	92	77	15
Shaped PVDF	92	72	20	92	75	17	92	72	20

Figure 35. Table of sound reductions.

SECTION 5.0

ACTUATORS

The early active structural acoustic investigations used point force actuators as control transducers. There are disadvantages of such devices, mainly that they are cumbersome and require some form of restraining back support or large reaction mass. In view of these limitations, much work has been carried out by VPI&SU in order to develop actuators bonded to or embedded in the structure itself.

5.1 PIEZOELECTRIC ACTUATORS

Piezoceramic actuators are noted for very high force and low displacement outputs, and simplicity in use. The most noted ceramic is Lead Zirconate Titanate (PZT), but the number of piezoelectric materials available is rapidly expanding. The U.S. Navy continues to sponsor most of the basic material research for sonar transducer and hydrophone applications. These unique ceramics are available in a variety of forms as stacks, sheets, laminates, and bimorphs, and with different directional sensitivities. They are attractive for high frequency applications where small displacements are effective.

VPI&SU has considerable experience in the application of piezoceramic actuators to active control of composite structures. This research program has developed a "generalized laminated plate theory" including induced strain, adhesive layers, and a "conservation of strain energy" analysis scheme for a laminate theory that includes spatially distributed induced strain actuators. These investigations have included the effects of adhesive layer thickness and stiffness on actuation authority. Most of the adaptive structure experimental work to date has dealt with the attachment (i.e., bonding) of PZT type actuators, but theoretical studies have been performed to gain a better understanding of the structural dynamic implications of embedding actuators.

Analytical work has shown that piezoceramics can be bonded to the structure surface and used to excite selected modes of vibration [31]. The piezoceramics can be tailored in shape and location in order to selectively excite certain modes of vibration. This has the advantage of reducing control spillover into unwanted structural or acoustic response. Figure 36 shows impressive results for control of sound radiation from a baffled circular panel of radius 0.2m with a small piezoceramic patch of dimensions $R_1 = 0.15\text{m}$, $R_2 = 0.18\text{m}$, $\theta_1 = -15^\circ$, $\theta_2 = 15^\circ$, bonded on the panel [32]. Figure 37 gives plots of panel displacement under the same conditions across the same diagonal as the radiation pattern. Interestingly, this result (and others) predict that the piezoceramic tends to force a node in the panel response near the actuator edge. This result is in line with the analysis of Ref. 31 which predicts that the piezoceramic effectively exerts a line moment on the structure (for pure bending cases) around its periphery. Thus the above results imply that a good location of the edge of a piezoceramic is near nodal lines and also show possibilities for edge control where structural rotation is high while

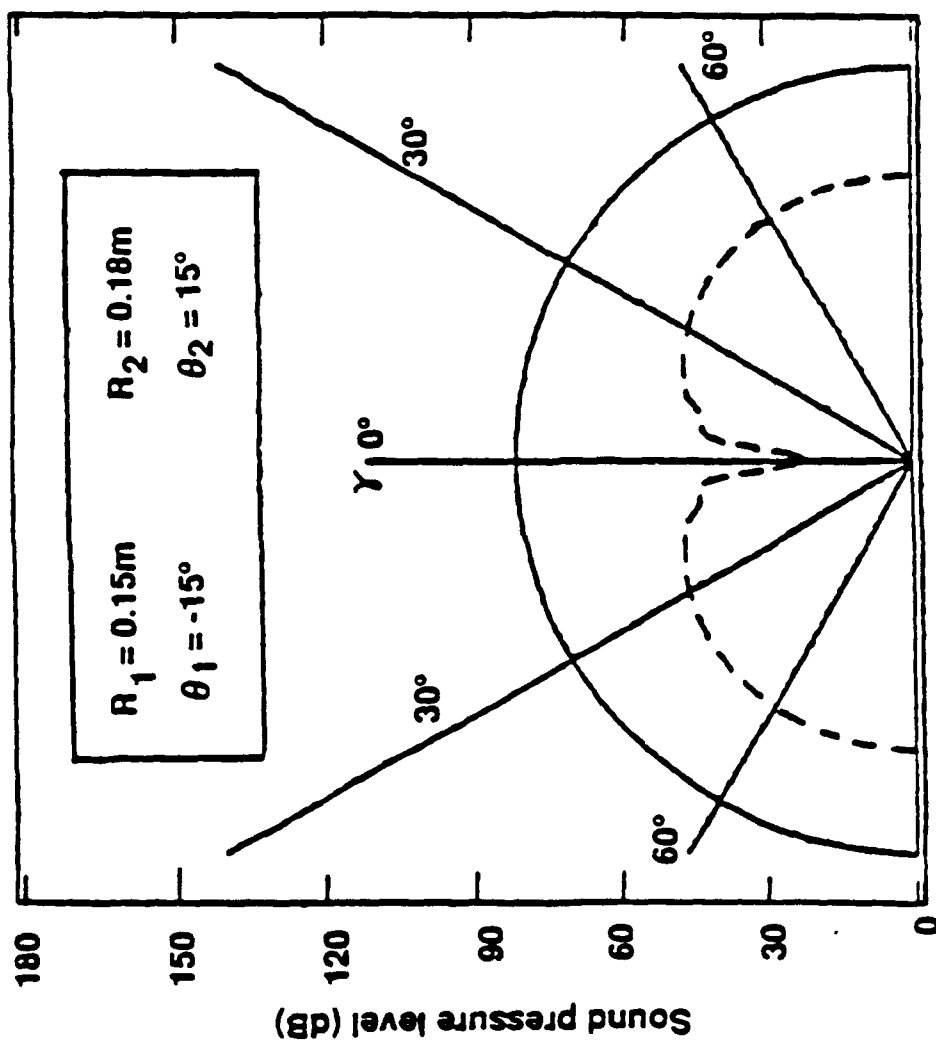


Figure 36. Radiation directivity. —, no control; ---, with control.

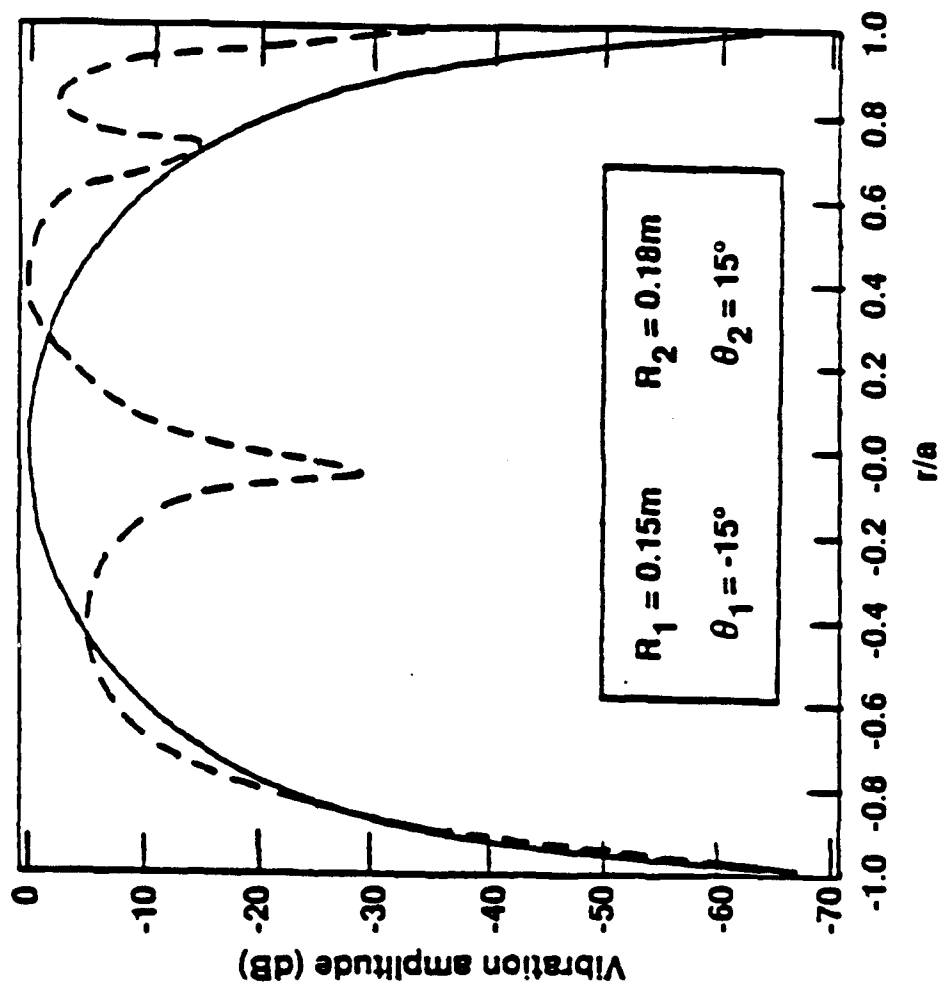


Figure 37. Panel displacement. —, no control; ---, with control.

out-of-plane vibrational motion is low. However, in general, the center of the transducer should be located in regions of high structural surface strain.

Figure 38 gives experimental results for radiation control from a baffled rectangular panel by a single, centrally located, small piezoceramic patch [33]. The panel has the same dimensions as that discussed in Figures 9 and 10 except it has a thickness of 2 mm. The panel is being driven near the resonance of a (3,1) mode at a frequency of 350 Hz and the error sensor is a single microphone located at 1.8 m from the panel. The results of Figure 38 show excellent global attenuation and panel modal decompositions demonstrate that all panel modes are significantly reduced in amplitude; a surprising result for a single actuator.

5.1.1 CLASSICAL LAMINATE PLATE THEORY (CLPT)

Classical laminated plate theory (CLPT) [34] has been applied to a laminated plate with induced strain actuators, such as piezoceramic patches [35]. The patches are bonded to its surface or embedded within the laminate, as illustrated in Figures 39 and 40 respectively, to develop an induced strain actuation theory that allows for the actuator patch to be spatially distributed. Several cases were examined including pure bending and pure extension. In addition, an angle-ply laminate with an embedded piezoceramic patch was addressed to show the coupling of bending and extension. The equivalent external forces and moments induced by the actuator patches can be determined so that the responses can be easily solved.

This classical laminate plate theory has been revised to include the use of the strain-energy model (SEM) which has been additionally developed for finite-length spatially distributed induced strain actuators for beams and plates [36]. A conservation of strain-energy model was developed by equating the applied moment on the cross section of the edges of the actuators to determine the induced linear strain distribution and equivalent axial force and bending moment induced by the actuators. In addition, more general conditions were included in this work; for example, multiple actuators can be embedded in any layer of laminate. This approach compares favorably with several other modeling approaches, specifically finite element (FEM) code [37] using generalized laminated plate theory (GLPT), and the spherical pure bending model (SPBM) [38] that was developed earlier in this research program. An application of this approach to examine is a simply supported beam. The beam is made up of steel with a length of 0.4 m and a thickness of 0.002 m with G-1195 piezoceramic patches attached to the top and bottom of the beam as shown in Figure 41. For this verification it is being harmonically excited in a pure bending manner. Figure 42 shows the modal amplitude distributions and the steady-state modal response for the piezoelectric actuators being driven at $\omega = 400$ rad/sec, a frequency between the first and second modes. Figure 43 shows the results for $\omega = 700$ rad/sec, a frequency very near the second mode. These comparisons show numerous advantages of the SEM over other models. Specifically, the SEM gives accurate results for a wide range of thickness ratios between actuators and beams, and has been extended to the two-dimensional problems, i.e., laminated plates. This research has gone further to include the development of closed-form solutions of the induced strain field from piezoelectric actuation.

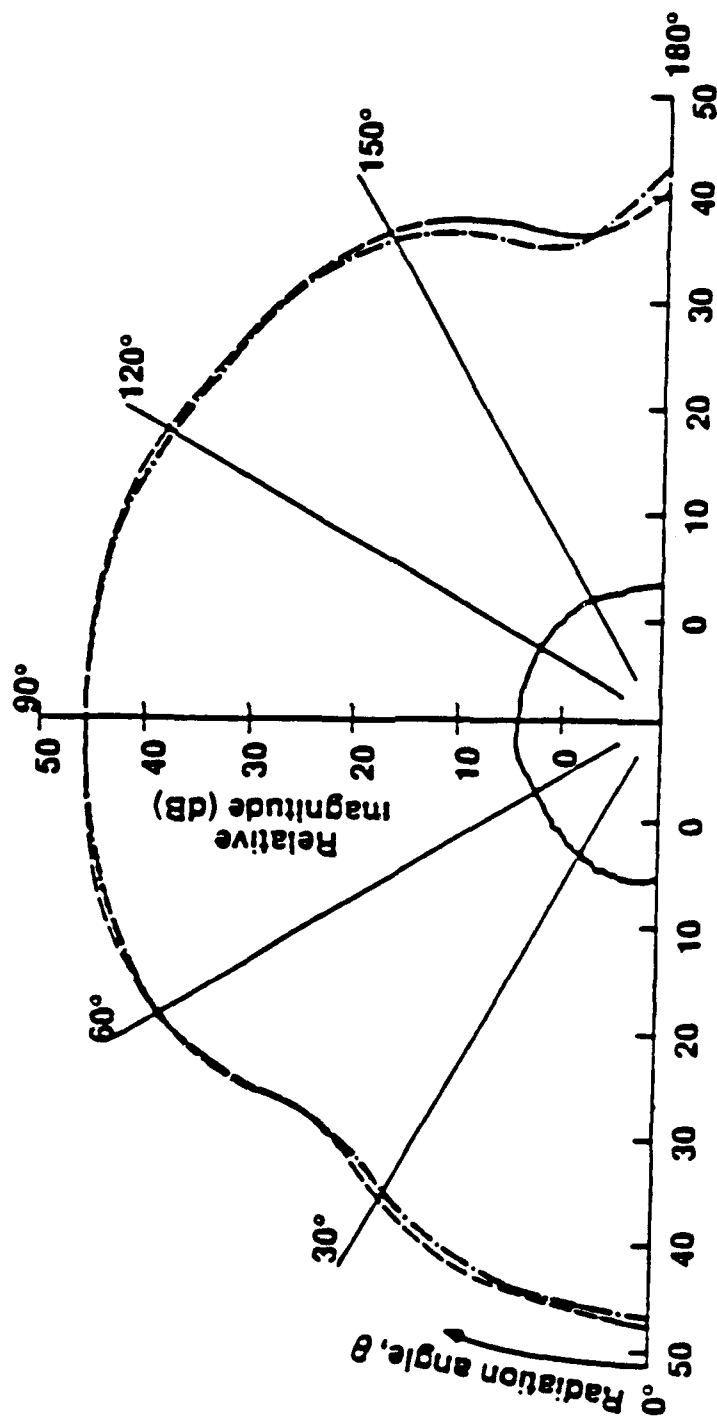
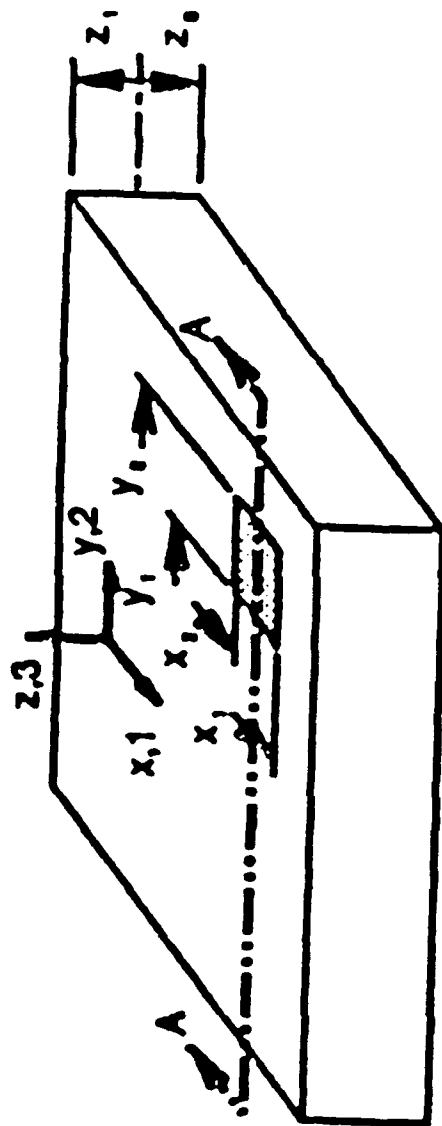


Figure 38. Radiation directivity. —: —, no control; - - - -, control field; —, with control.



AA Section

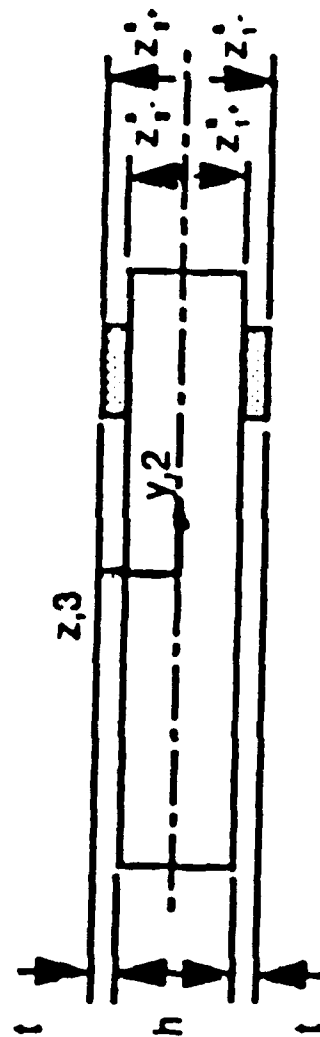
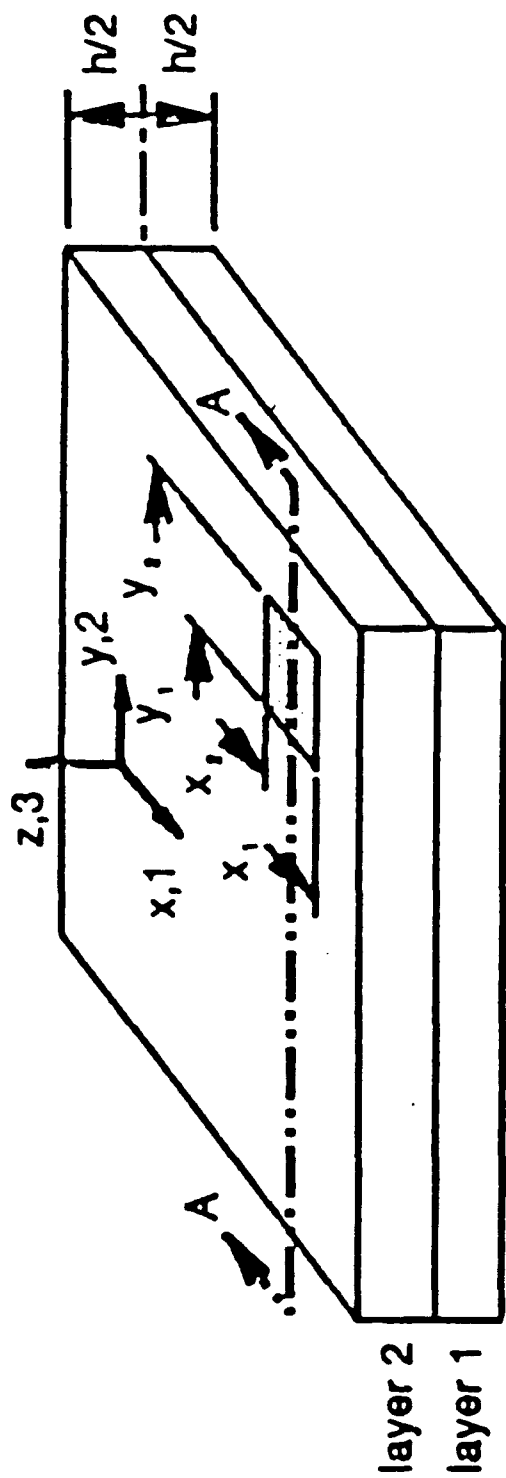


Figure 39. Bonded piezoceramic actuators.



AA Section

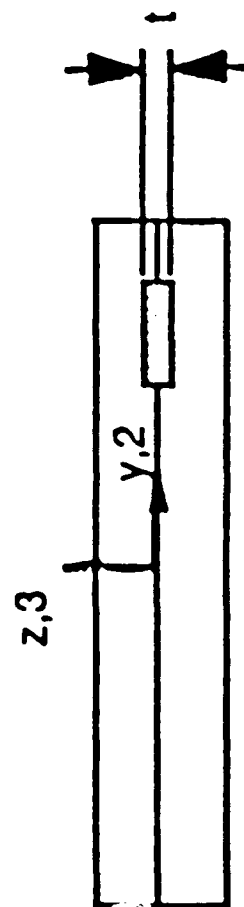


Figure 40. Imbedded piezoceramic actuator.

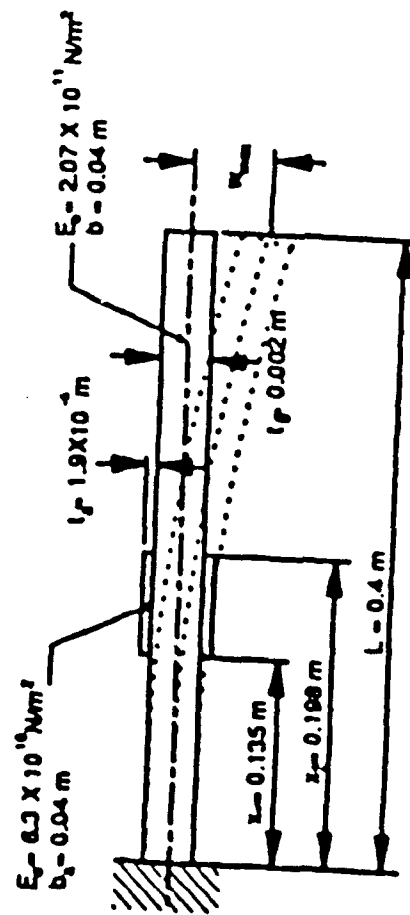


Figure 41. Beam with bonded piezoceramic patches.

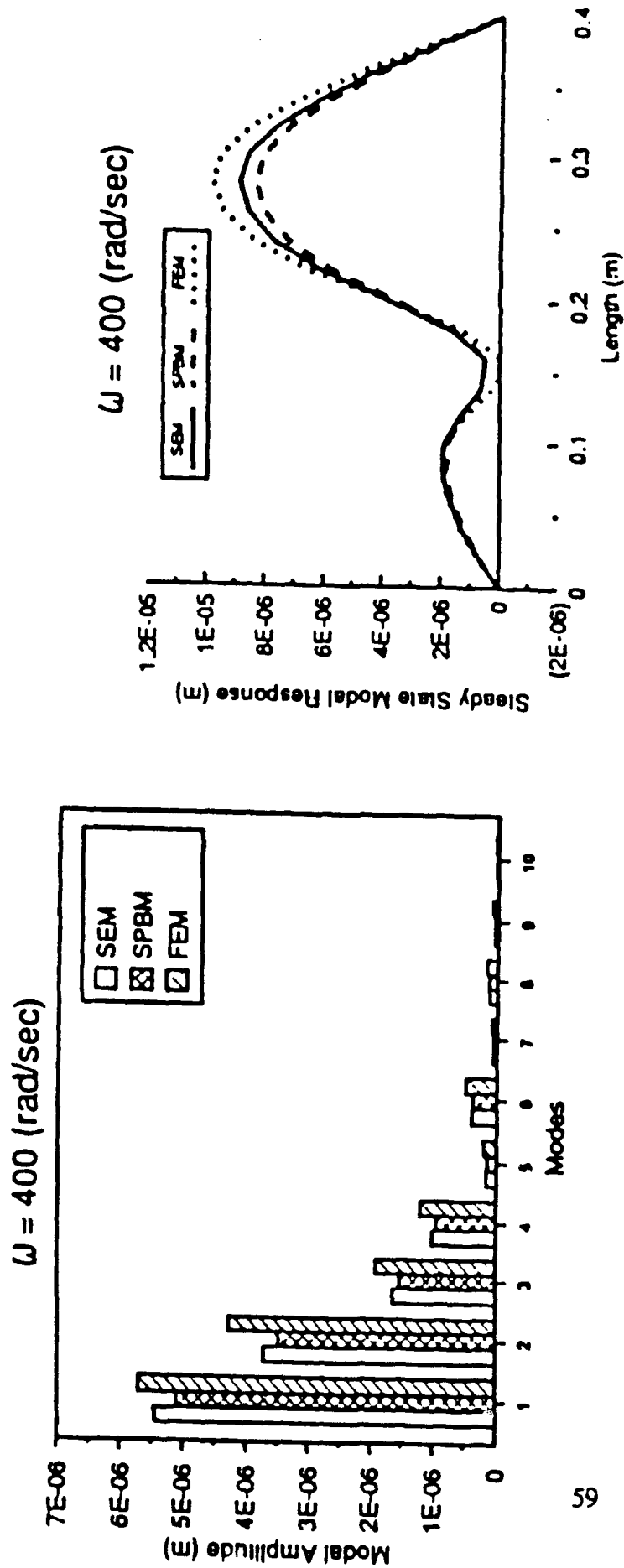


Figure 42. Modal amplitude and modal response for $t_b/t_s = 10.49$.

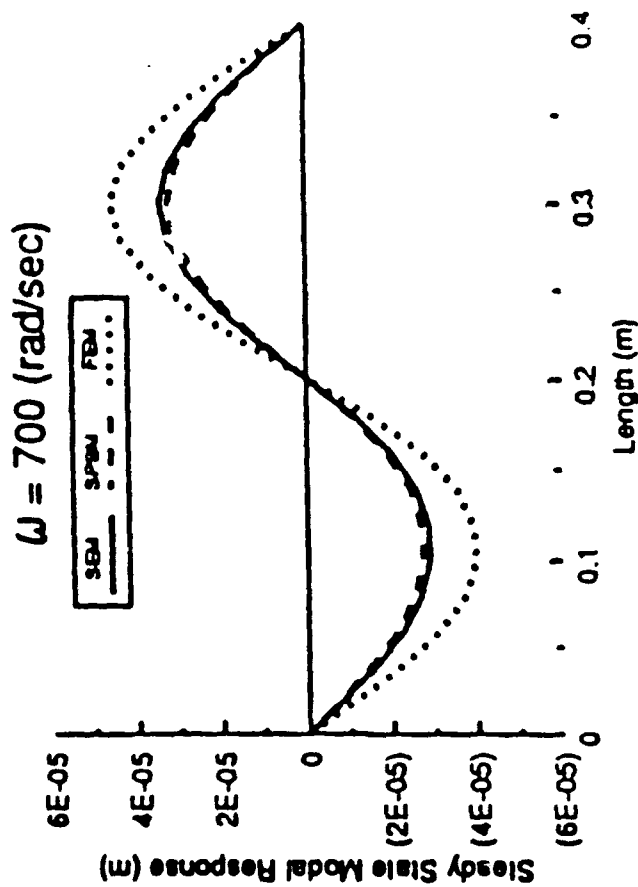
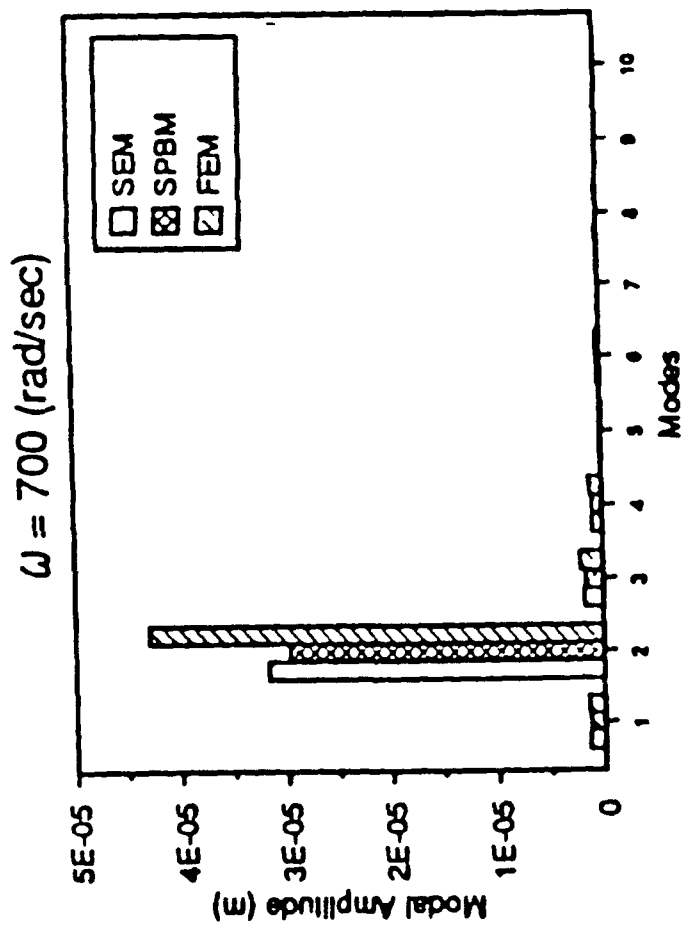


Figure 43. Modal amplitude and modal response for $t_b/t_a = 10.49$.

5.1.2 PVDF ACTUATORS

PolyVinylidene Fluoride (2) (PVDF 2) thin films are most noted as sensors (as fully discussed later in this report), but in recent years they are gaining increased attention as actuators for activation of light flexible structures. VPI&SU is investigating the development of distributed PVDF 2 structural acoustic actuators for structures that are curved where it would be difficult or impractical to employ piezoceramic patches for one or more reasons.

5.2 SHAPE MEMORY ALLOY ACTUATORS

As early as 1938 the shape memory effect had been observed in brass, but it wasn't until W. J. Buehler and R. C. Wiley in 1962 [39] at the then U.S. Naval Ordnance Laboratory discovered the alloy Nitinol (Nickel-Titanium Naval Ordnance Laboratory) that provided a radical and practical memory effect. Very recent research has produced numerous other shape memory materials such as NiTiCu, NiTiPd, and CuZnAl, and including plastics. Nitinol is still the most frequently used alloy with a number of improvements, and it is an outstanding memory material that can provide on the order of 100,000 psi of recovery force. Shape memory alloy (SMA) provides a very compact, light weight, and highly reliable means of actuation, but the response times are relatively slow since it is a thermal process.

The basic property is that these alloys undergo a reversible Martensitic-Austenitic crystal phase transformation. A mechanism design process begins with the alloy being shaped into a desired form at high temperature, and then cooled while maintaining the desired form. At normal low temperature (martensitic condition), when plastically deformed it will retain the shape as an ordinary material. Then with the external stresses removed it will regain its original (memory) shape when heated to a transition temperature (much less than the original forming temperature). The process is the result of a martensitic phase transformation taking place during heating with considerable energy back to its original set form.

Other important work in this area has been concerned with the use of shape memory alloys (SMA) as adaptive inputs. The main aim of the work is the development of adaptive structures utilizing embedded shape memory alloys. The applications of such "smart" structures are many [40] and much work has been carried out to derive the fundamental equations of motion of such structures [41]. It should be noted here that, in contrast to the piezoceramic transducer, the active input is not oscillating and what is really being developed here is a truly adaptive (or smart) structure whose properties and response can be altered by electrical inputs. Thus, this particular implementation represents a semi-active control approach. Work is, however, progressing on developing a vibrating SMA actuator that would allow full active control approaches to be implemented. However, the discussion here will be limited to those topics related to the structural acoustic problem, and the research accomplished under this Grant.

5.2.1 CONSTITUTIVE MODELS

Figure 44 shows the typical nonlinear coupling that exists between the recovery stress, initial plastic strain, and the activation temperature. However, constitutive models have lacked several important fundamental concepts that are essential for many proposed intelligent material system applications such as shape memory hybrid composites. A complete unified, one dimensional model of shape memory materials has been developed [42]. The thermomechanical model investigations have incorporated important material characteristics involved in the internal phase transformation of shape memory alloys. These characteristics include energy dissipation in the material that governs the damping behavior, stress-strain-temperature relations for pseudoelasticity, and the shape memory effect. Based upon this one-way effect model a two-way model has been further developed [43]. This model reflects the essence of SMA in a very simple form, and therefore is easy to use in engineering designs. The material parameters required by this model can be calculated or measured using standard material testing apparatus. The computer simulation results are in agreement with the experimental data that has been published in the open literature. It has been demonstrated that developed theory can quantitatively predict and describe the behavior of shape memory alloys and be imposed on the two-way shape memory effect.

A multi-dimensional thermomechanical constitutive model of SMA has been additionally developed [44]. The constitutive relation is based upon a combination of both micromechanics and macromechanics. A variable of martensite fraction is introduced in this model to reflect the martensitic transformation that determines the unique characteristics of shape memory alloys. This constitutive relation can be used to study the behavior associated with 2-D and 3-D SMA structures.

5.2.2 SMA HYBRID COMPOSITES

VPI&SU developed the novel techniques termed "Active Property Tuning (APT)" and "Active Strain Energy Tuning (ASET)" [41] with SMA fibers embedded in hybrid composites, or even bonded in such a way that the material can be stiffened or controlled by the addition of heat (i.e., apply a current through the fibers). The SMA fibers are placed in or on the structure in such a way that when activated there is no resulting deflections, but instead the structure is placed in a 'residual' state of strain. The resulting stored energy (tension or compression) changes the energy balance of the structure and modifies the modal response. Figure 45 shows the active modification of mode shapes for a $[0^\circ/-45^\circ/45^\circ/90^\circ]$, SMA-graphite epoxy hybrid laminate controlled by active strain energy tuning (ASET) [45]. For the case in which 90-deg laminate are activated, the mode shapes change considerably from the inactivated plate. The fourth mode shape of the 90-deg ply activated is essentially the 10th mode shape of the inactivated plate. The ability to change the mode shape of a structure has significant impact on the control of structural acoustic radiation. For example, the third mode of the 90 deg activated plate has a higher radiation efficiency than the third mode of the inactivated plate, which will result in a considerably different transmission loss profile.

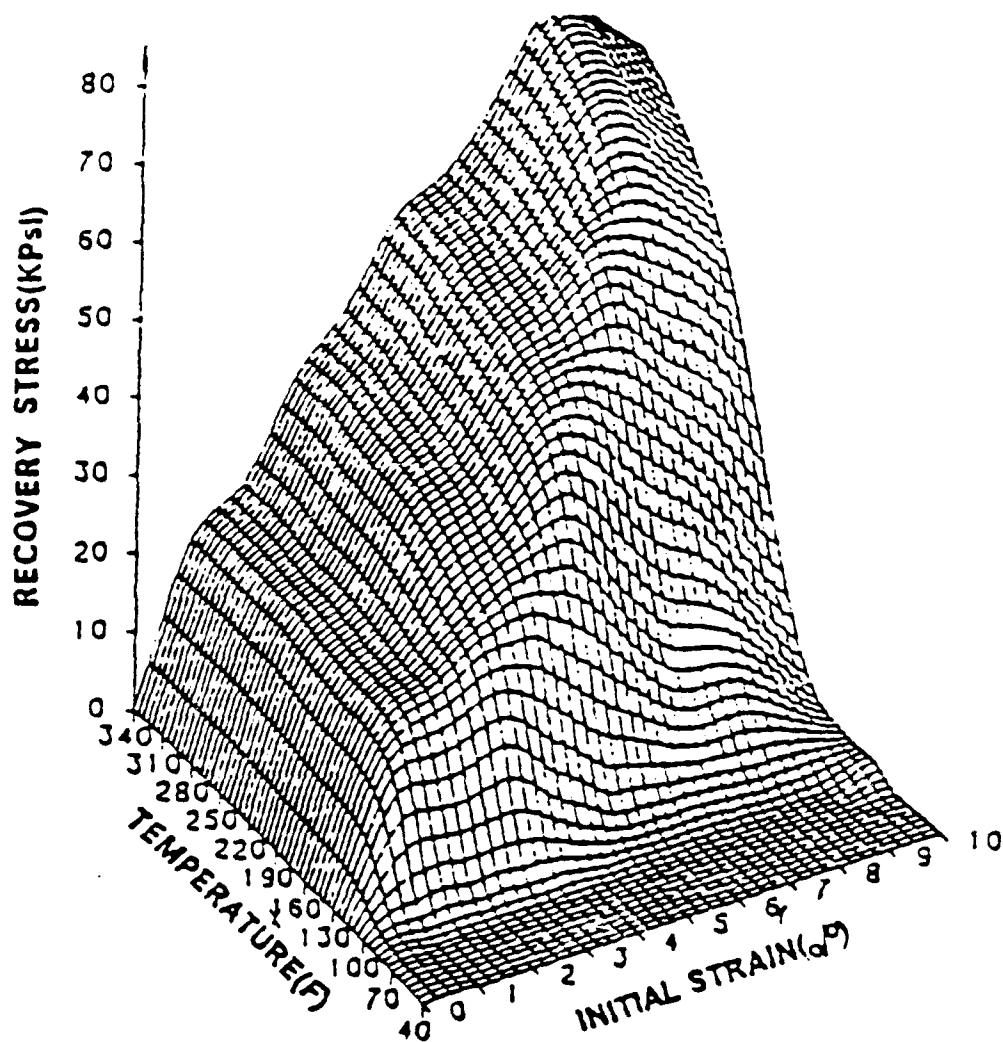


Figure 44. Stress-strain-activation temperature coupling of Nitinol.































First Ten Mode Shape of a Quasi-isotropic Plate			
All Activated Inactivated	90° Activated	45° Activated	
 1	 1	 1	
 2	 2	 2	
 3	 3	 3	
 4	 4	 4	
 5	 5	 5	
 6	 6	 6	
 7	 7	 7	
 8	 8	 8	
 9	 9	 9	
 10	 10	 10	

Figure 45. Comparison of mode shapes of a [+45°/-45°/0°/90°] square plate using ASET.

Obviously, if the geometric mode shapes of a composite can be modified, the natural resonant frequencies of the structure can also be significantly altered. Figure 46 shows a comparison of the first natural frequency of a clamped-clamped graphite-epoxy beam as a function of temperature and a SMA graphite-epoxy hybrid beam [46]. The SMA hybrid beam has Nitinol volume fraction of 15%. The first natural frequency of the SMA hybrid beam increases from 21 Hz at room temperature to 62 Hz when the actuators are heated to 300 °F (149 °C). This type of adaptive structural modification also has significant implications on active structural acoustic control. The previous results show adaptive changes in mode shape for all except the higher order modes. For this motion a different configuration has been developed.

Sound radiation has been demonstrated to be related to modal response and shape [1]. Thus if one can alter the mode shapes of a vibrating structure, then the acoustic power radiated by such structures may be reduced (or increased). Analytical work on simply supported panels with embedded SMA fibers of various configurations has shown that when the fibers are activated (i.e., heated by passing a current through them) then significant modification in the radiated acoustic field is achieved. Figure 47 shows radiation directivity for an activated SMA panel and activation of the fiber can be seen to cause a significant change in the radiation pattern [47]. This effect has been demonstrated to be due to changes in the panel mode shapes and thus the structural acoustic coupling when the fibers are activated [48]. For the analytical example of Figure 47, the composite plate was of dimension 1.1 m x 0.8 m x 0.008 m, with a quasi-isotropic stacking sequence of $[0^\circ/-45^\circ/45^\circ/90^\circ]$ and constructed from equal thickness layers of graphite/epoxy and Nitinol/epoxy. The Nitinol was assumed to be the top and bottom plys of the laminate with a 40% fiber volume percentage, yielding 10% for the entire plate. The Nitinol was also assumed to achieve a recovery stress in the fibers upon activation of 280 MPa. The damping factor was set equal to 0.01.

Figure 48 presents transmission loss (TL) curves for sound transmitting through a SMA panel with and without activation [47] and excited by a plane wave incident at 45°. The main point demonstrated in Figure 48 is that significant modification in the TL curve can be achieved by SMA. Much work is now needed to optimize the layout and activation of the SMA fibers to optimally reduce sound radiation in a required way, based on formulations for the modification in modal shapes and their corresponding radiation efficiency.

Figure 49 shows a number of activation scenarios and their first four mode shapes for an analytical plate case study for more complex SMA fiber configurations [49]. These results are based upon an analytical modeling technique which uses the Ritz method, classical laminated plate theory, and finite panel acoustic radiation theory to predict the modal and structural acoustic behavior of locally activated SMA hybrid composite panels. This information is further used to determine the transmission loss versus frequency as shown in Figure 50.

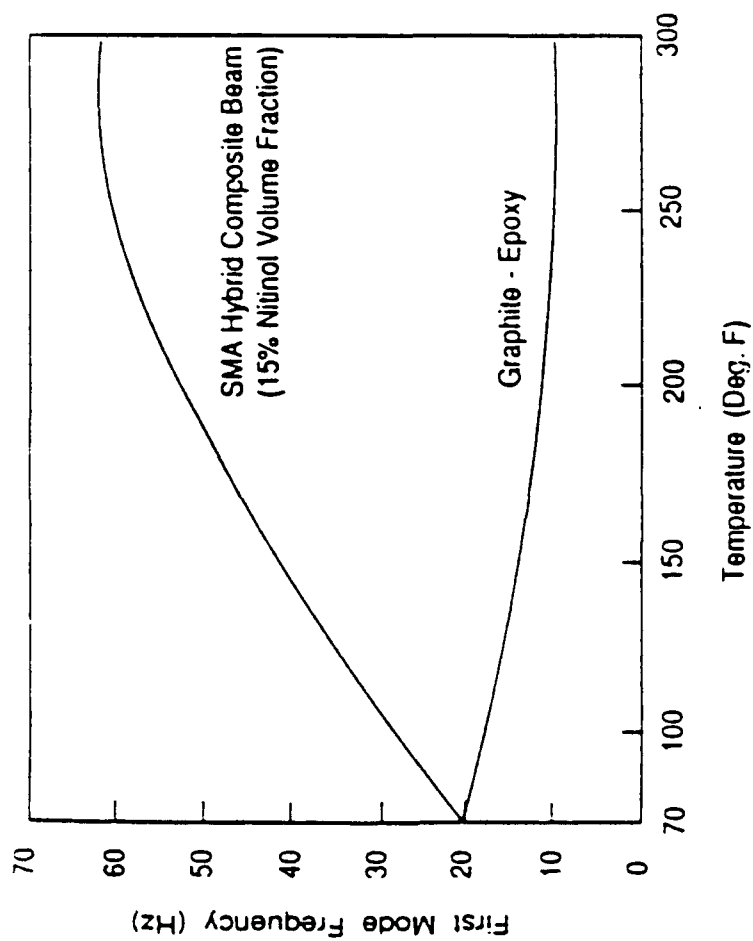


Figure 46. Comparison of SMA hybrid beam and graphite-epoxy beam.

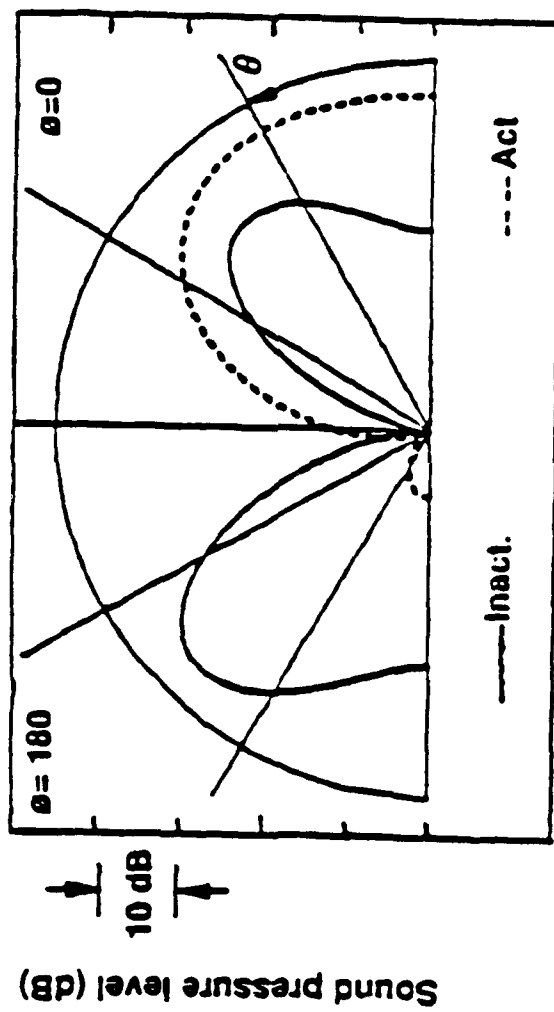


Figure 47. Radiation directivity.

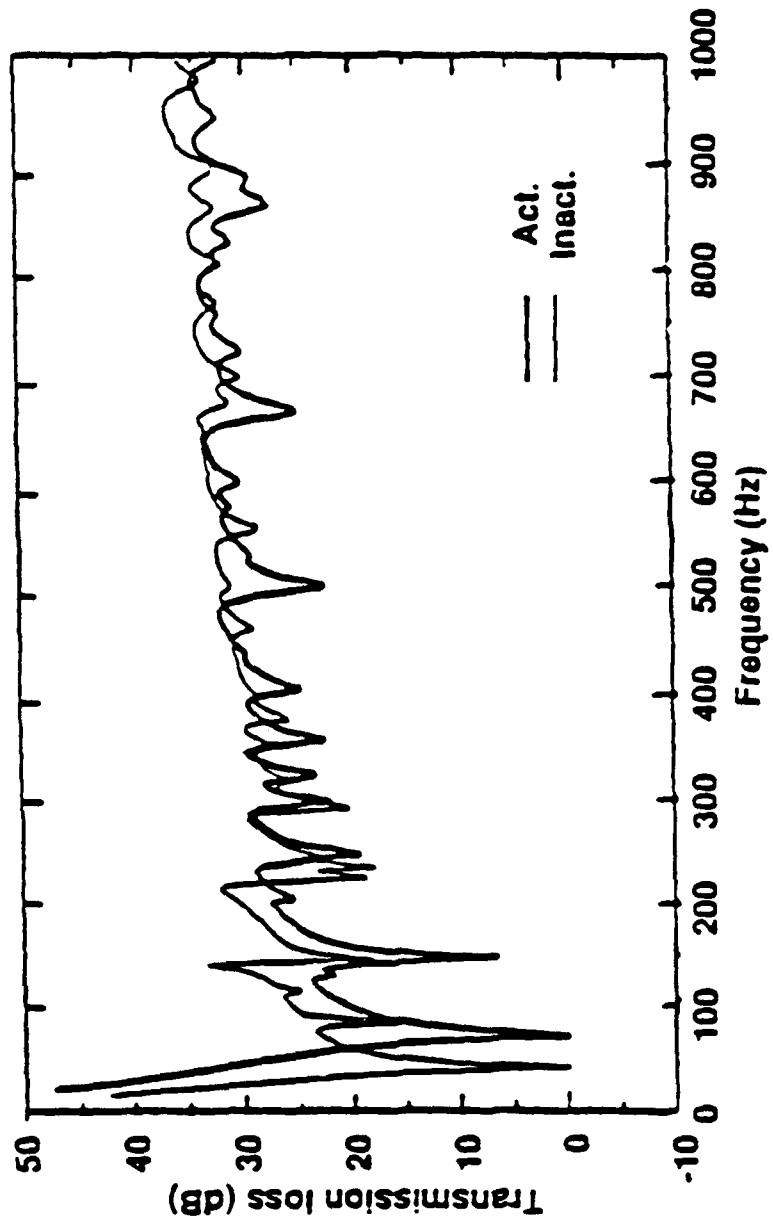


Figure 48. Panel transmission loss.

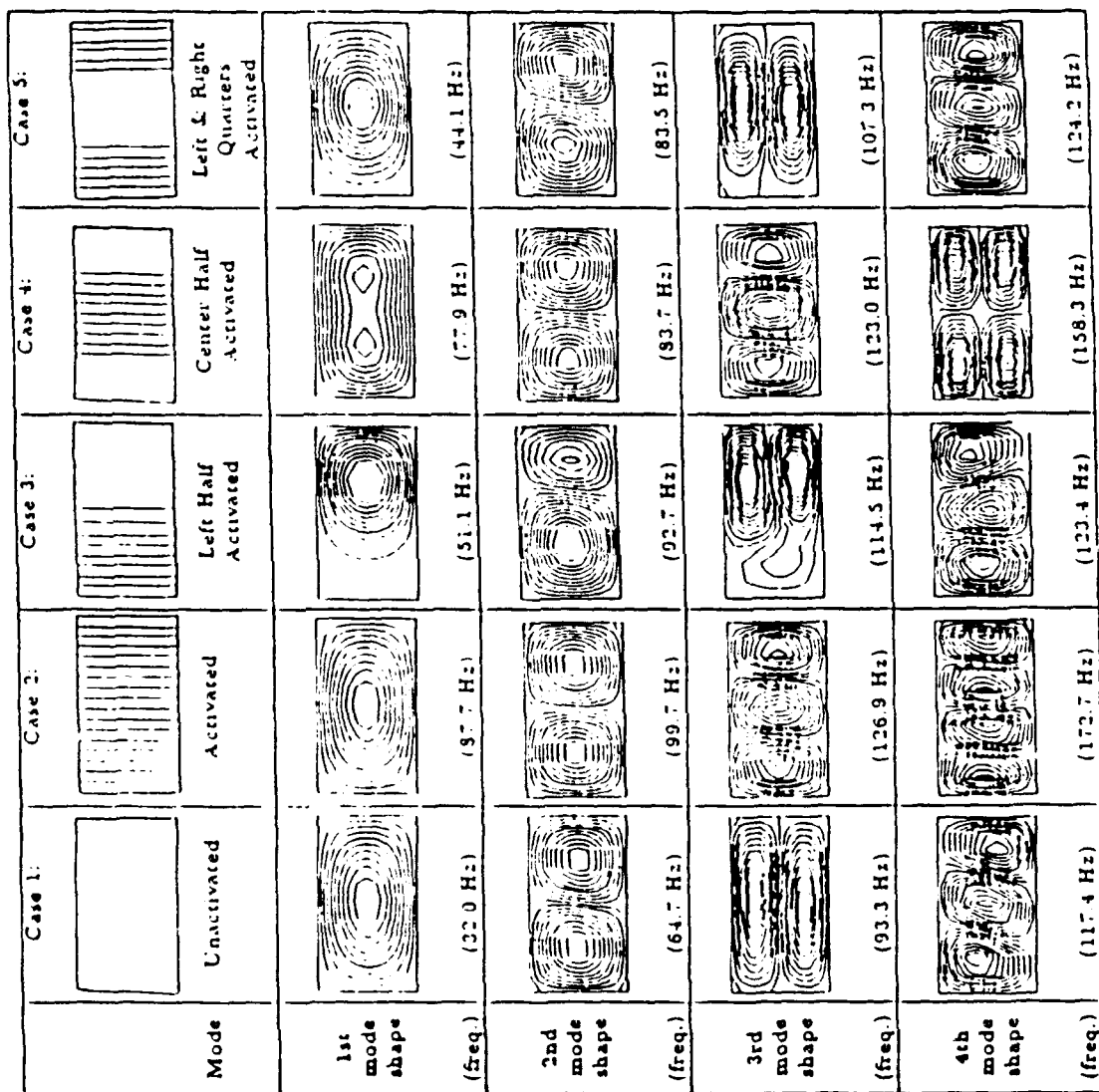


Figure 49. Panel activation scenarios and mode shapes for 1st case study.

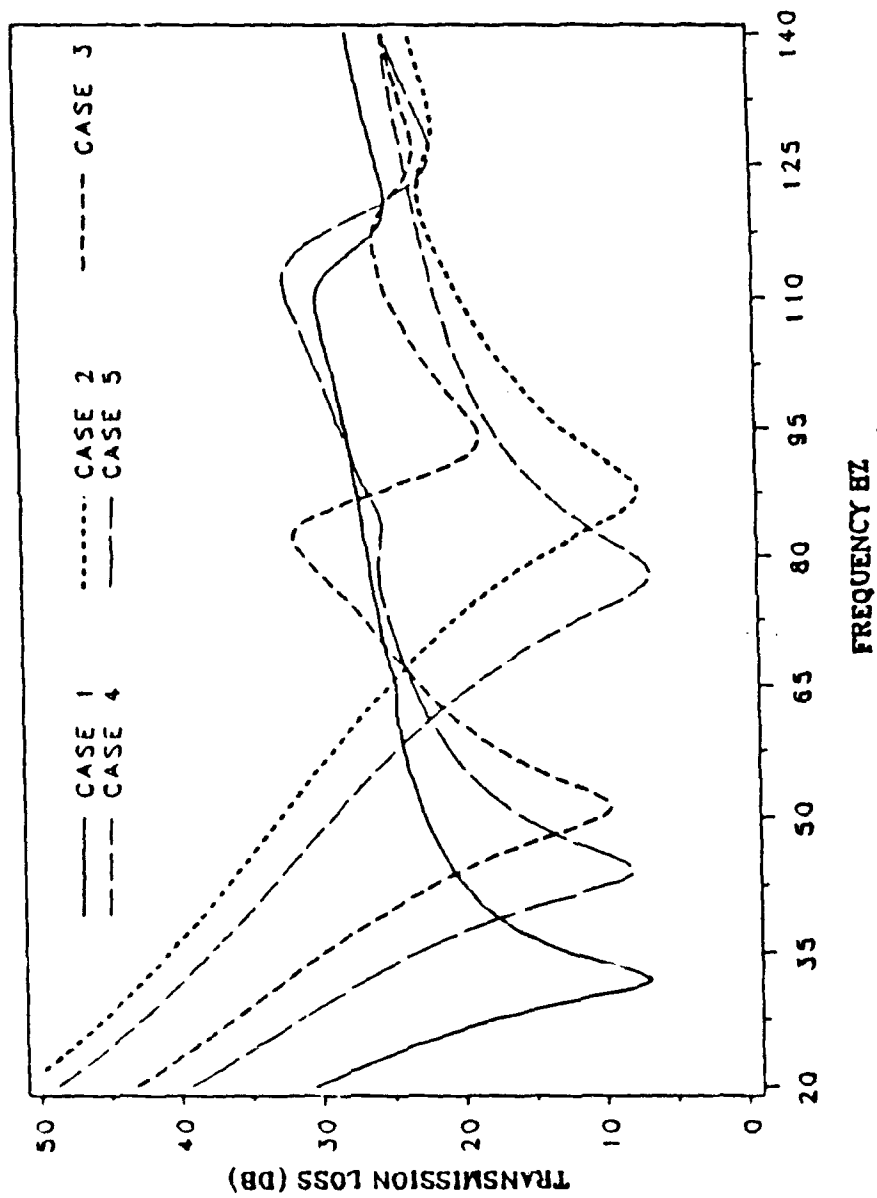


Figure 50. Transmission loss vs. frequency.

5.2.3 ALTERNATE RESONANCE TUNING (ART)

The concept of alternate resonance tuning (ART) is when adjacent sound-transmitting panels in a structure are tuned to resonate above and below a particular frequency such that they oscillate and radiate sound with equal source strength and opposite phase, incident sound is readily attenuated by the structure at and near the frequency to which the panels are tuned [50]. A simply supported shape memory (SMA) hybrid composite panel can be locally stiffened by selectively activating embedded SMA fibers to achieve ART adaptively, thereby enabling the panel to attenuate sound over several activation-dependent frequency ranges.

For example, Figure 51 shows a 0.2 m wide band of nitinol fibers being activated such that the structure is essentially divided into three substructures: two relatively large compliant sections with a stiffener section between them to provide an activation-induced pseudo-rib or boundary [51]. The SMA fibers in this middle section alter the structure to resemble two adjacent panels with pseudo-support or stiffener at their interface. Modal analysis results for the first two resonances, along with on-resonance transmission loss for panels with l_1/l_2 ratios of 1.0, 0.9, 0.8, and 0.7, are presented in tabular form in Figure 52, and Figure 53 presents representative transmission loss versus frequency characteristics..

5.2.4 CONTROLLED BEAM RADIATION

Other work of interest concerns control of sound radiation from beams with embedded SMA fibers. Figure 54 shows the shift in measured resonant frequency of the fundamental mode of a clamped-clamped beam upon SMA activation [52]. Experiments on controlling radiation from such beams using SMA fibers have also been successful [53]. In this experimental arrangement, the beam has dimensions of 0.822 m, 0.206 m x 0.001 m and was constructed from a Nitinol reinforced graphite-epoxy material with a 15% Nitinol volume fraction. The beam was excited into motion by a non-contacting electromagnetic transducer driven by a steady state pure tone frequency. The beam could thus be excited either on or off resonance of required modes by varying the input frequency. Figure 55 shows the time history of the sound pressure level of the beam responding to an excitation at the $n=4$ mode (145 Hz). The control strategy uses a pattern search technique, implemented on a PC/AT computer, to minimize the sound pressure, by adjusting the voltage applied to the embedded SMA fibers. For this case, the effect of the third mode moving close to the excitation frequency is evident at 40 seconds. The controller then adapted the input so that the disturbance was located at the minimum pressure response (an anti-resonance point) between the third and fourth modes.

While the focus of this research effort has been the development of SMA actuators embedded in hybrid composites, other SMA actuator forms have been investigated (such as springs) that can be effectively employed in many actively controlled structural acoustic systems [54]. It is not always required that the SMA actuator be embedded, and there can be some significant advantages for not doing so. A natural concern about embedding SMA in a composite is the impact upon the strength and reliability of the composite, and these issues have been under investigation [55].

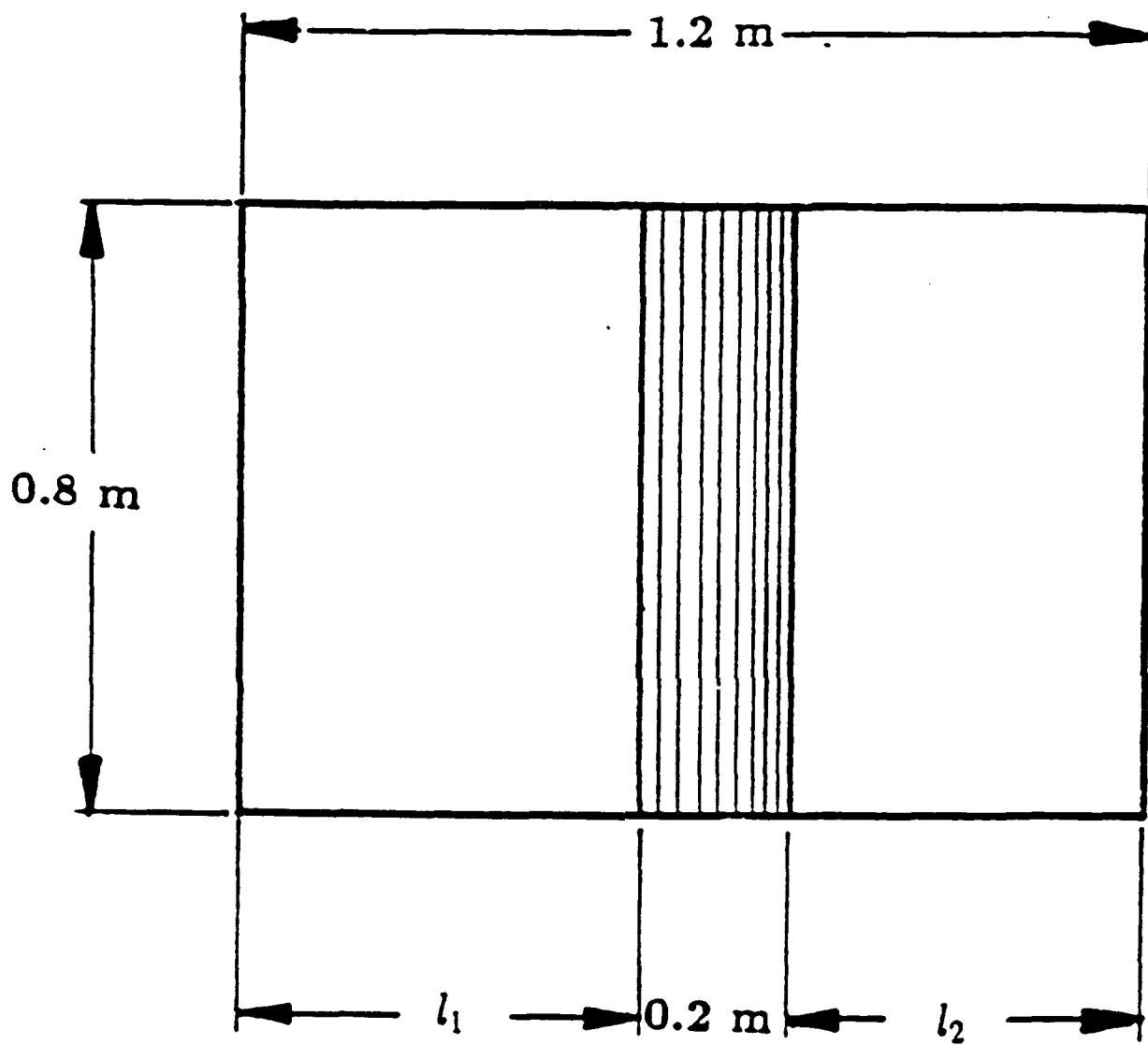


Figure 51. Panel activation geometry.

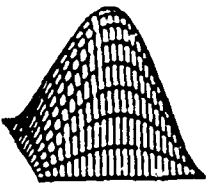
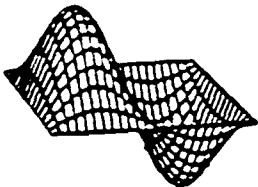
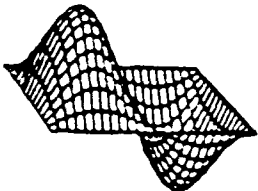
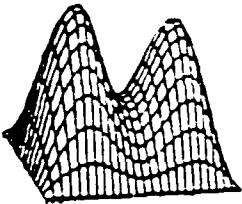
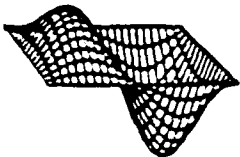
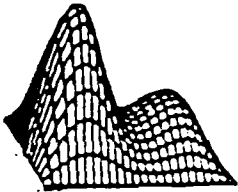

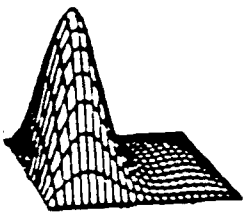
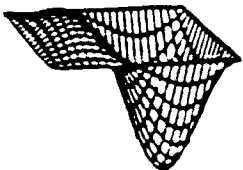
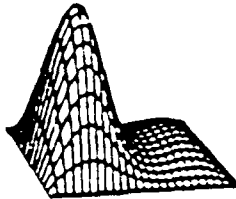
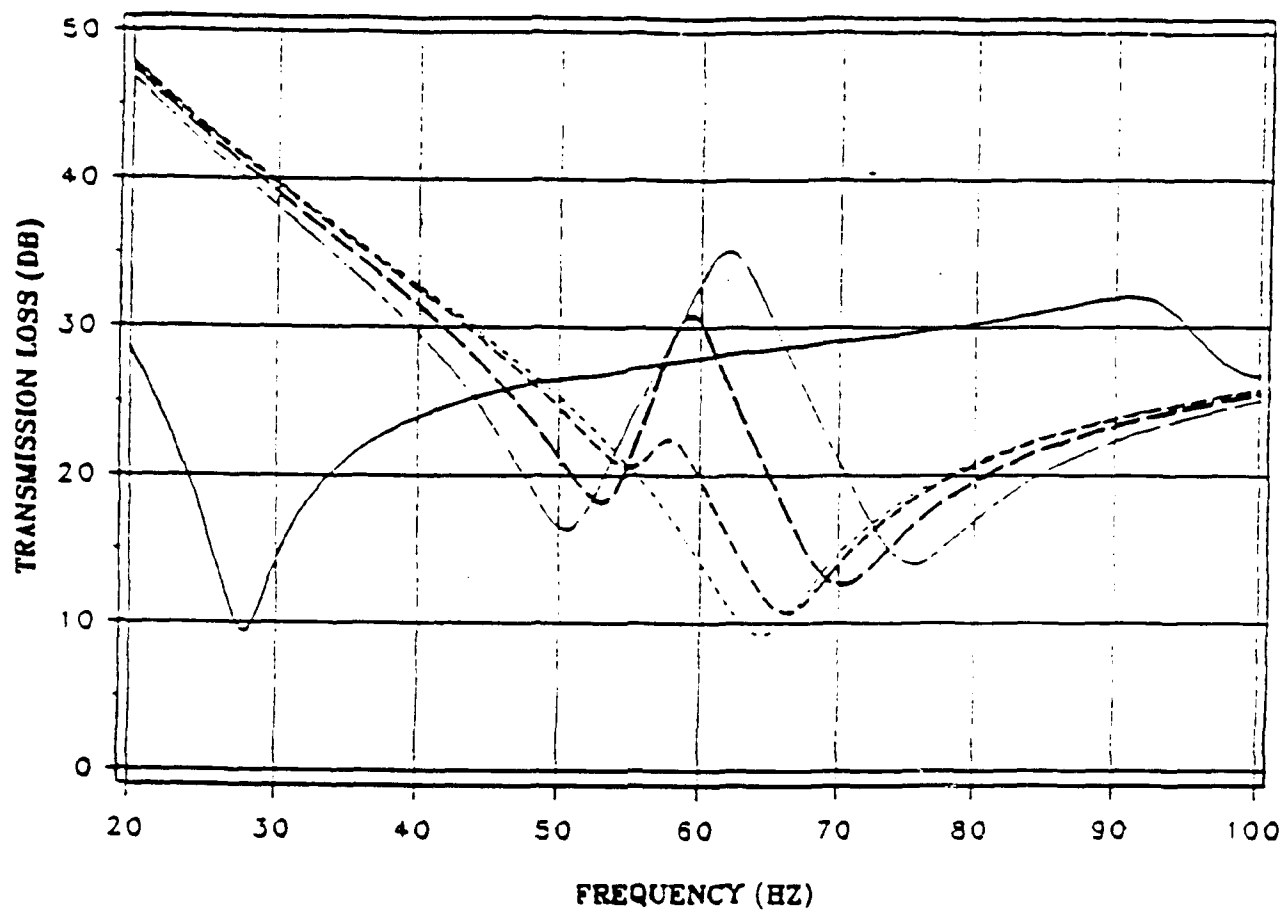
ACTIVATION	FIRST MODE SHAPE	FREQUENCY (TL)	SECOND MODE SHAPE	FREQUENCY (TL)
Unactivated		27.7 Hz (9.3 dB)		52.6 Hz (26.8 dB)
$\frac{f_1}{f_2} = 1.0$		57.3 Hz (17.8 dB)		64.3 Hz (9.2 dB)
$\frac{f_1}{f_2} = 0.9$		56.2 Hz (21.2 dB)		65.9 Hz (10.7 dB)
$\frac{f_1}{f_2} = 0.8$		53.6 Hz (18.5 dB)		69.6 Hz (12.6 dB)
$\frac{f_1}{f_2} = 0.7$		50.8 Hz (16.3 dB)		75.0 Hz (14.0 dB)

Figure 52. Modal analysis summary.



ACTIVATION: ——— UNACTIVATED L1/L2 = 1.0
 - - - - - L1/L2 = 0.9 - - - - - L1/L2 = 0.8
 — — — L1/L2 = 0.7

Figure 53. Transmission loss vs. frequency.

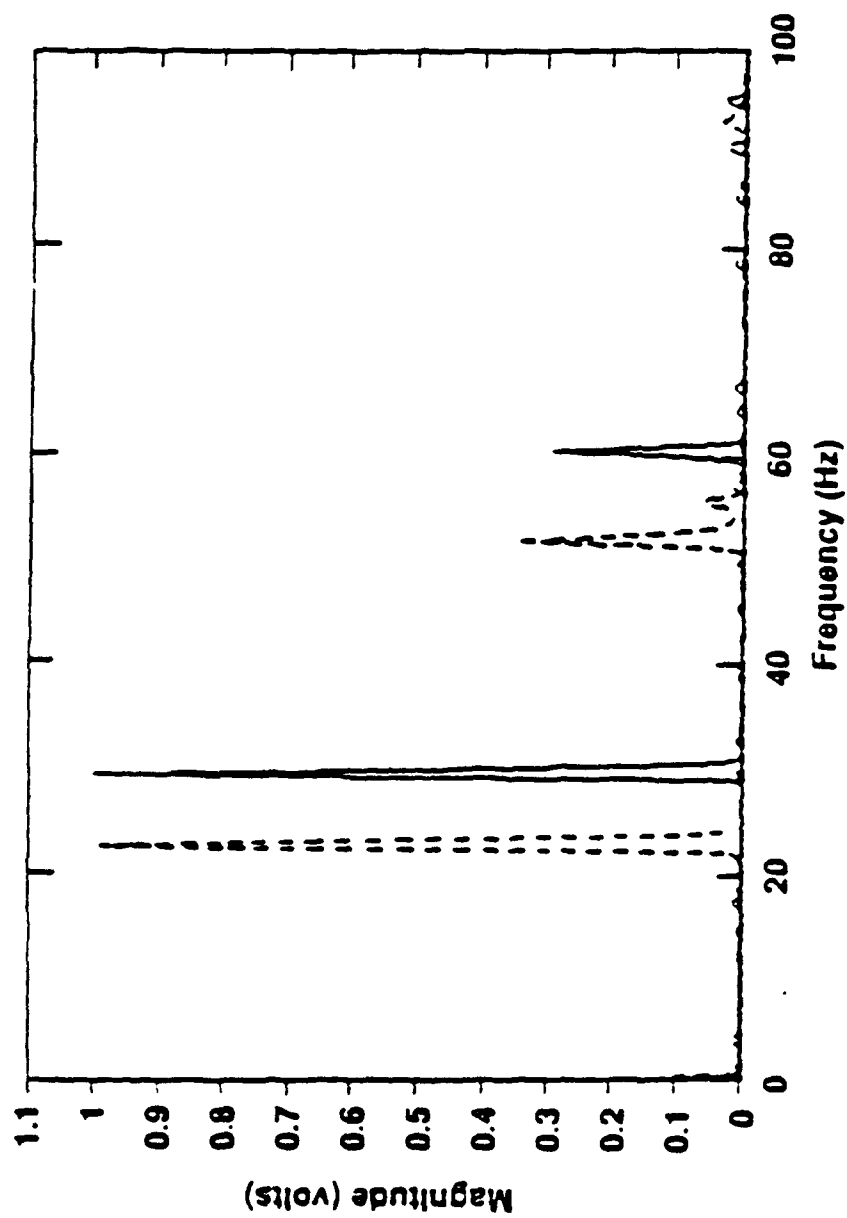


Figure 54. Adaptive SMA beam response. —, unactivated; —, activated.

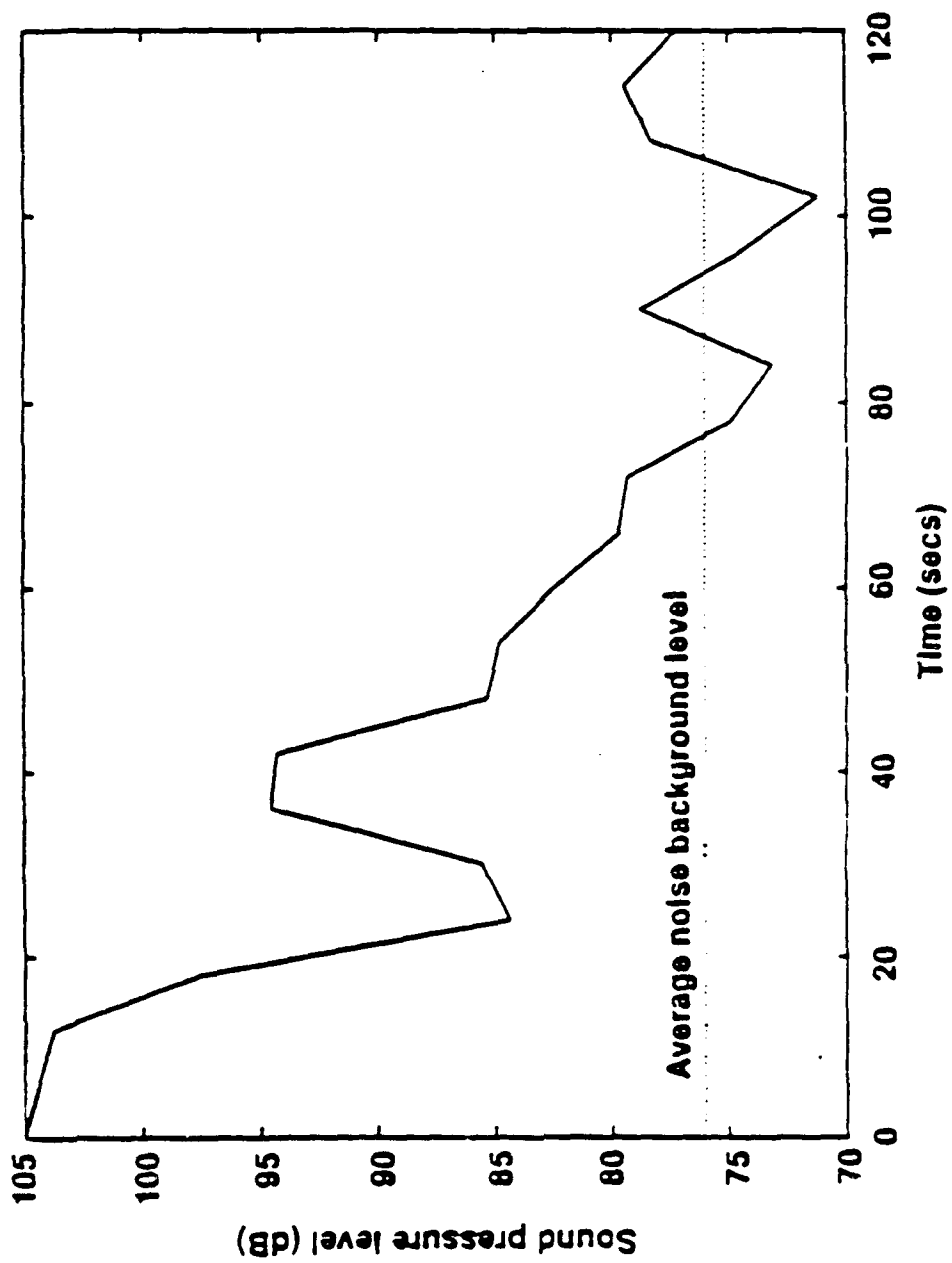


Figure 55 Adaptive control of radiation from SMA beam.

SECTION 6.0

SENSORS

For any control strategy, observability is an important requirement. Thus sensing of the plant responses to be minimized is an important part of the research work described here. As stated previously, a key aspect of the approach is to use the radiated acoustic field as direct error information. Most of the previous results discussed here relate to using discrete microphones located in the radiated acoustic field. However, there are many situations where such an approach is impractical and alternatives have to be sought.

Composite materials have advanced as a basic structural material, and it provides both an opportunity and in some cases the requirement for integrating unique sensors. This has led to the concept of smart structures that can perform structural health monitoring during fabrication, construction, and operational use, and realtime sensory acquisition and processing for structural behavior, loads, and other sensory information about the internal and external environment. Sensors can be attached by bonding or even embedded during the original fabrication process. VPI&SU has been at the leading edge in these innovations and developments.

6.1 OPTICAL FIBER SENSORS

Optical fiber waveguides, originally developed for medium distance data communication, have been applied to the sensing of physical observables for the past ten years [56]. By monitoring the intensity, phase, polarization, wavelength, mode and time delay properties of optical signals which propagate in such fiber waveguides, a wide range of relevant physical phenomena to the problem here including strain, rotation rate, vibrational mode shape amplitudes and acoustic waves may be measured.

Optical fiber sensors have properties which make them advantageous in certain applications. The sensor consists of a very small single strand of glass which can be embedded in, or attached to, a structure with minimal loading [57]. Signals from fibers optic sensors neither contribute to nor respond to electromagnetic noise. This property becomes important in systems with dense electronic equipment, or when electromagnetic shielding is difficult. Fiber optic sensors have excellent sensitivity and extremely high bandwidth so they can be used to detect vibrations over a wide range of frequencies. In addition they can be designed to operate under a variety of environmental conditions including high temperatures, high pressure, and harsh chemical concentration [58 and 56].

Research under this Grant has contributed to recent advances in the development of practical modal domain (MD) fiber optic sensor implementations which include lead-in/lead-out insensitive fibers and elliptical core sensing fiber. This work also entails integration of such approaches into the structural acoustic control problem. Figure 56 gives a spectral estimation of the response of a beam measured by an accelerometer and an embedded optic fiber [59]. It is apparent that

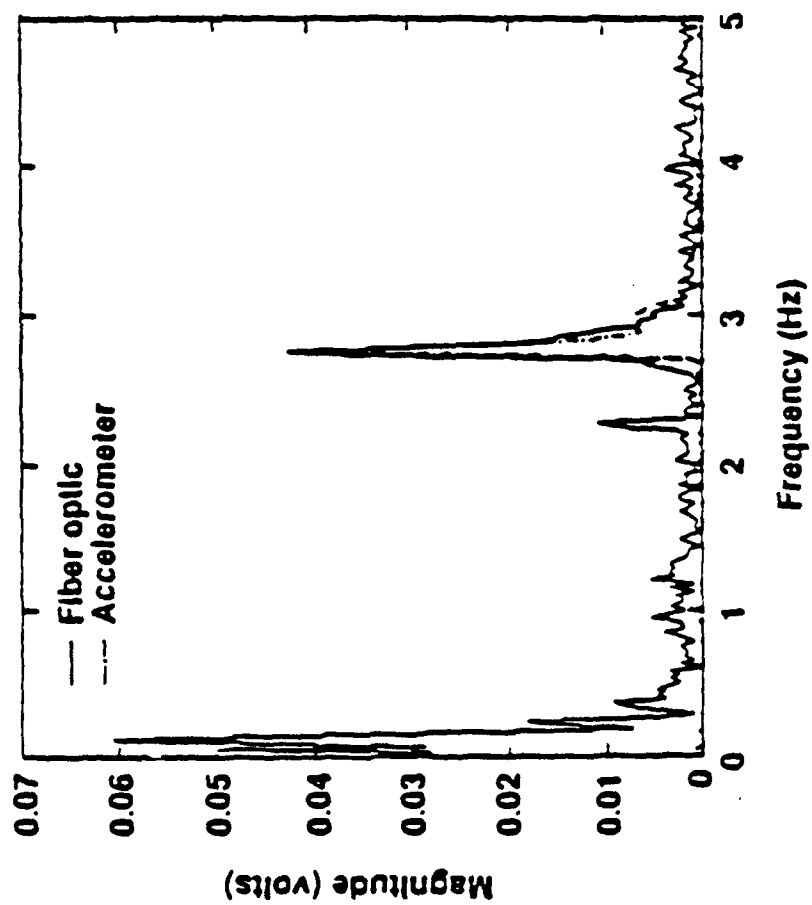


Figure 56. Fiber optic sensing of beam response.

the optic fiber gives a good estimation of the resonance frequency of the beam. In other applications, optic fibers have been used in conjunction with Kalman filtering to provide modal amplitudes of beam responses required for implementation of control [60]. Figure 57 gives results of a simulation of such a system on a clamped-clamped beam when excited by a steady state sine wave turned on at $t=0$. There is no damping in the system. The curves are the actual (r_1) and estimated amplitudes (\hat{r}_1) of mode 1. The results show that the modal amplitudes can be estimated accurately with a time domain approach. In order to derive a control variable in the radiated acoustic power these modal amplitudes may be weighted by their associated radiation efficiency, thus artificially introducing the structural acoustic coupling characteristic. To fully account for radiated power, the modal cross interaction terms must be considered as well. However, future work will be concerned with direct sensing of near-field acoustic pressures using optic fibers. In conjunction with acoustic holography theory [61] it will then be possible to construct a control cost function in terms of estimated far-field radiated acoustic power, from near-field measurements of pressure.

6.1.1 OPTICAL FIBER ASAC EXPERIMENT

VPI&SU has demonstrated for the first time active structural acoustic control using optical fiber sensors [62]. The fiber sensors were oriented symmetrically about the vertical centerline of a thin, simply-supported baffle plate and configured for most effective detection of the odd-odd plate modes as shown in Figure 58. The experiments were performed in the VPI&SU anechoic chamber, and the experimental interconnection configuration is illustrated in Figure 59. The output from the fiber sensor is used as an error signal, a least mean square (LMS) algorithm is used for digital processing and control is achieved using a single piezoelectric actuator. The sensor consists of an elliptical-core (e-core) fiber as the lead-in fiber and a circular core single mode fiber as the lead-out. Linearly polarized light is launched parallel to the major axis of the ellipse of the single mode lead-in fiber. The polarization preserving properties of the e-core fiber make this fiber section relatively insensitive to external perturbations. Details of the sensor construction have been previously described in Reference 63. The piezoelectric actuator consisted of two piezoelectric elements bonded symmetrically (front and back). A narrowband filtered-x version of the adaptive LMS algorithm is implemented on a TMS320C25 DSP board in an AT computer. For all the tests, the noise input to the plate was a shaker using a steady state sinusoid which served as the reference signal for the controller as well. Preliminary results obtained from the use of elliptical-core, two-mode fiber sensors attached to the plate show effective acoustic attenuation of 25 dB in on-resonance cases and 10 dB in off-resonance cases. The on-resonance test case results are shown in Figure 60 for the excitation frequency of 88 Hz, corresponding to the (1,1) mode of the plate. The residual modes were attenuated as well as seen in Figure 60 which represents what has been termed "modal reduction", since all modes were observed to be attenuated. The off-resonance test case for the frequency of 349 Hz, which lies between the (2,2) and (3,1) modes of the plate the results are shown in Figure 61. The modal response in this case is more complex, as can be seen in this figure. Before applying control the (3,1) mode is dominant in response; however, significant response is observed in the (1,1), (2,1), (2,2), and (2,3) modes of the structure. Upon achieving control all modes were

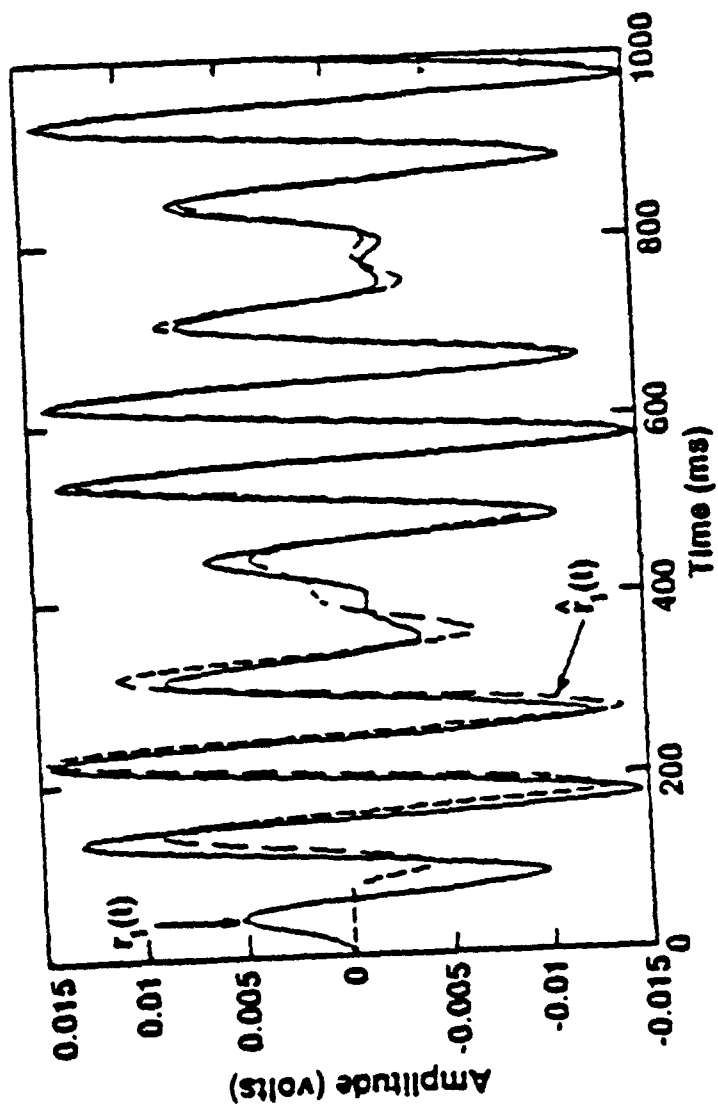
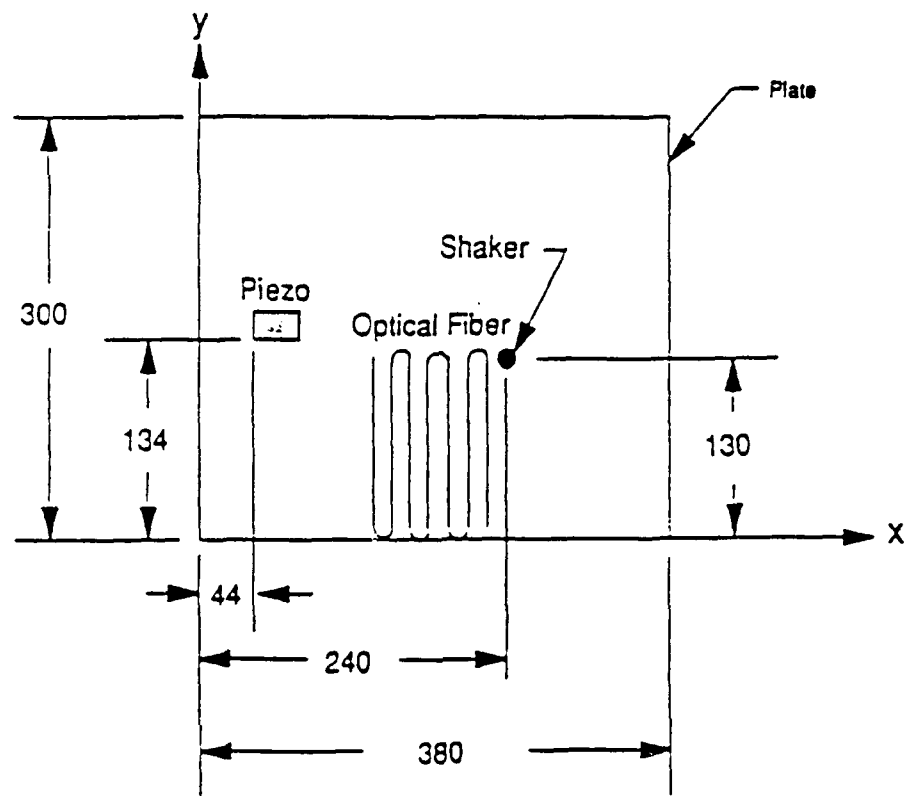


Figure 57. Kalman filtering modal identification.



All Dimensions in Millimeters

Figure 58. Schematic of plate with actuator, disturbance and sensor.

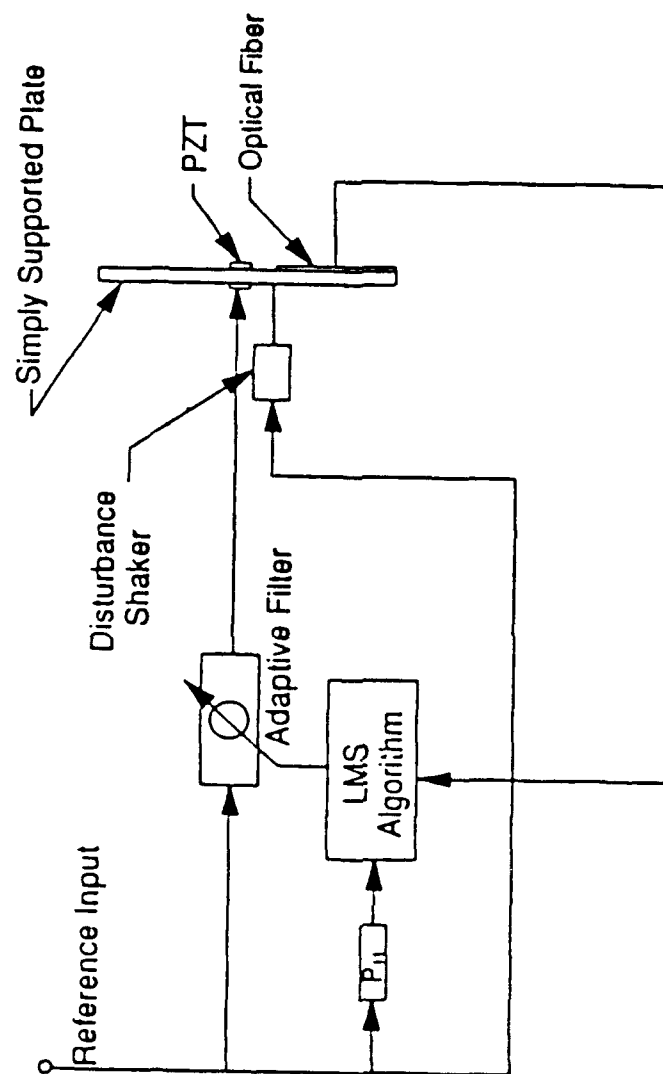
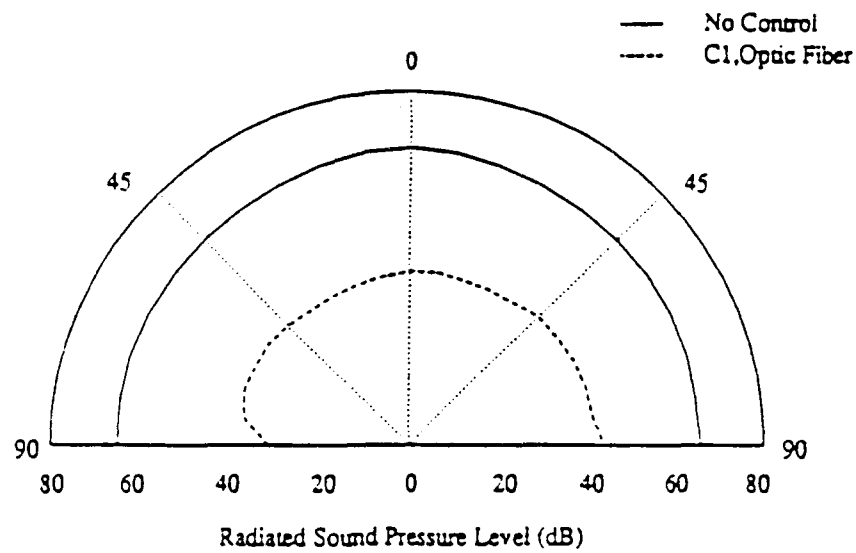


Figure 59. Schematic of control approach (single channel).



(a) Acoustic Directivity Pattern of Plate/Baffle System

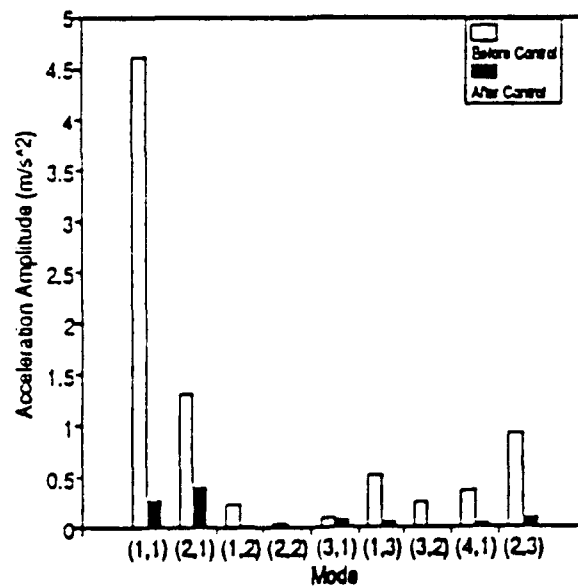
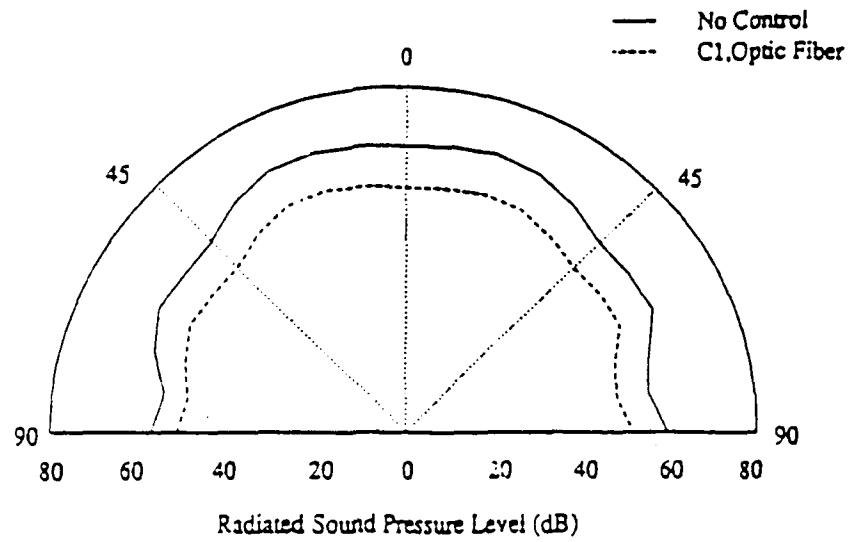


Figure 60. Excitation of (1, 1) mode at 88 Hz.



(a) Acoustic Directivity Pattern of Plate/Baffle System

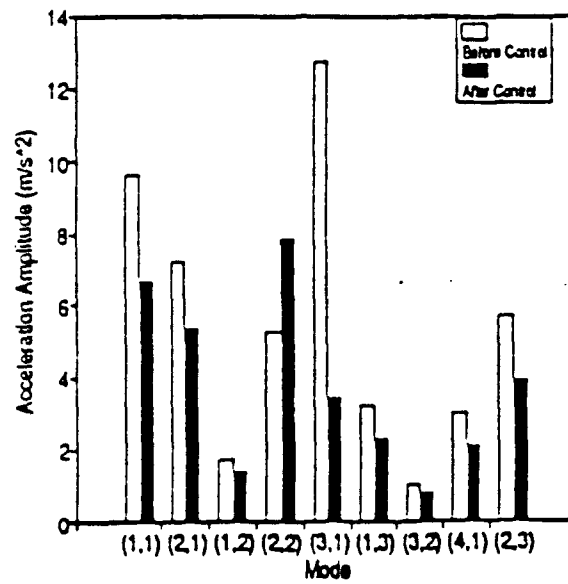


Figure 61. Off-resonance excitation at 349 Hz.

attenuated except mode (2,2), and thus this case is termed "modal restructuring", since the relative modal response was rearranged as opposed to being reduced.

6.2 NITINOL FIBER SENSORS

Nickel-Titanium shape memory alloy have many peculiar properties, many of which can be exploited for activation or sensing applications. As is the case with many activator materials, such as piezoceramics and PVDF, nitinol can be used to perform sensing as well as actuation. Nitinol strain sensors however differ from nitinol actuators in that the sensors utilize only one material phase, the austenite phase, whereas nitinol actuators utilize the reversible transformation between the martensitic and austenite phase. The distributed (or integrated) nitinol sensors currently being utilized measure strain. The utility of the integrated strain information has been discussed above in reference to the optical fiber sensors.

The nitinol fiber strain sensors are simply nitinol wires with a low transition temperature so as to assure that the material is always in the austenite phase. In our studies, pseudoelastic or super-elastic nitinol has been used. The basic concept is to measure the change in resistance of the nitinol as a function of integrated strain. This concept allows for very simple processing as the nitinol sensor is nothing more than a distributed strain gage in a Wheatstone bridge. Nitinol has a high resistivity for a metal, making it well suited for strain sensing. The super-elastic nature of the nitinol also means that strains up to 6% can be reliably and repeatedly measured.

Experimental demonstration of the nitinol sensor was performed by embedding a 0.012 in. diameter fiber in a fiberglass cantilever beam off the neutral axis. The nitinol sensor was then used as the active leg of a Wheatstone bridge. When the embedded nitinol fiber is strained, the resistance increases and the bridge is no longer balanced resulting in a voltage across the bridge. The cantilever beam was used to verify the integrated strain capabilities of the fiber in both static and dynamic modes. Static and dynamic tests showed a linear response for root strains (of the fiber) up to 1.2%, the maximum strain tested with the beam. Calibration tests of the nitinol fiber itself has indicated a linear response greater than 6% strain. However, once the sensing fiber has been strained beyond 6% it becomes plastically deformed. The experimental response of the embedded nitinol fiber to a freely vibrating cantilever beam at approximately 4 Hz is shown in Figure 62. The primary advantages of nitinol strain gages is their ease of implementation and large dynamic range. Current work involves using the nitinol fiber as a dual-mode sensor, i.e. to measure temperature and strain simultaneously.

6.3 PIEZOCERAMICS AND PVDF

Piezoceramics can also be used as sensors. The advantage of this strategy is the same devices can be used in some applications in a time sharing mode as actuators and sensors. Figure 63 shows a comparison of a FFT of the response of a panel excited by an impulse [33]. The response is measured by a centrally co-located accelerometer and a piezoceramic element bonded to the panel. The estimates of panel resonant frequencies by the piezoceramic can be seen to be good and thus the information could use as a control variable such as an estimate of the

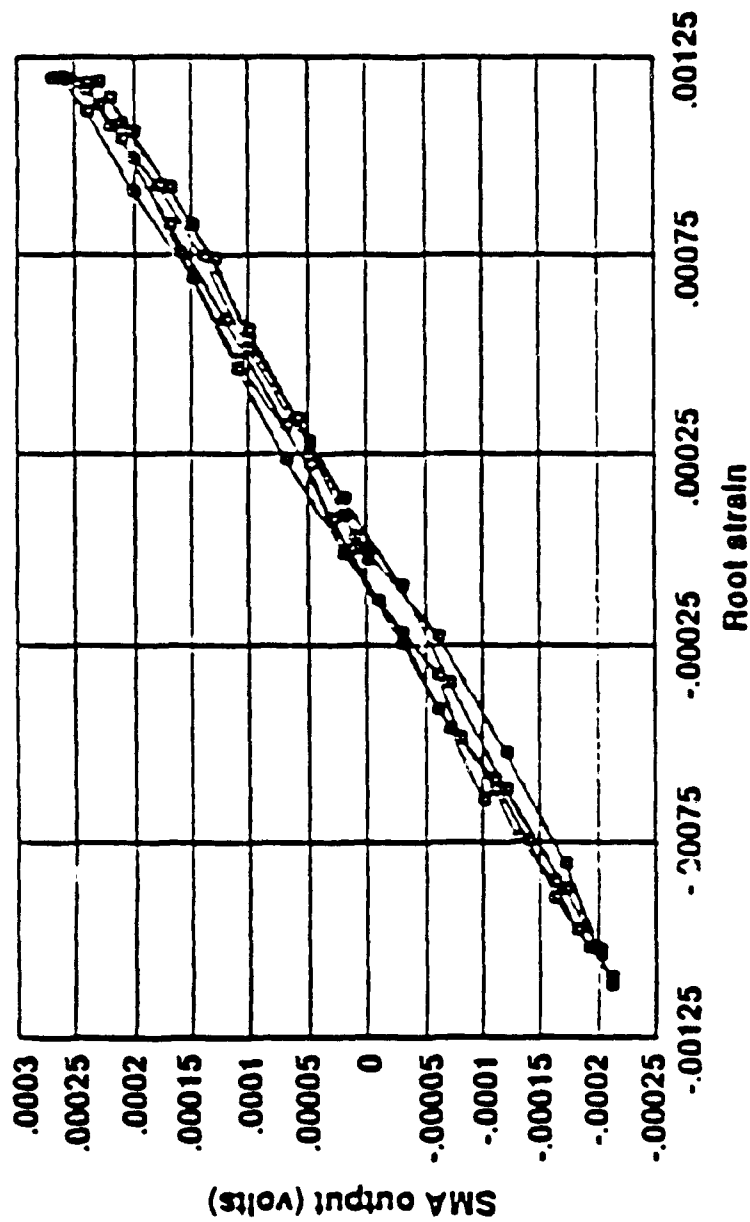


Figure 62. SMA strain sensor.

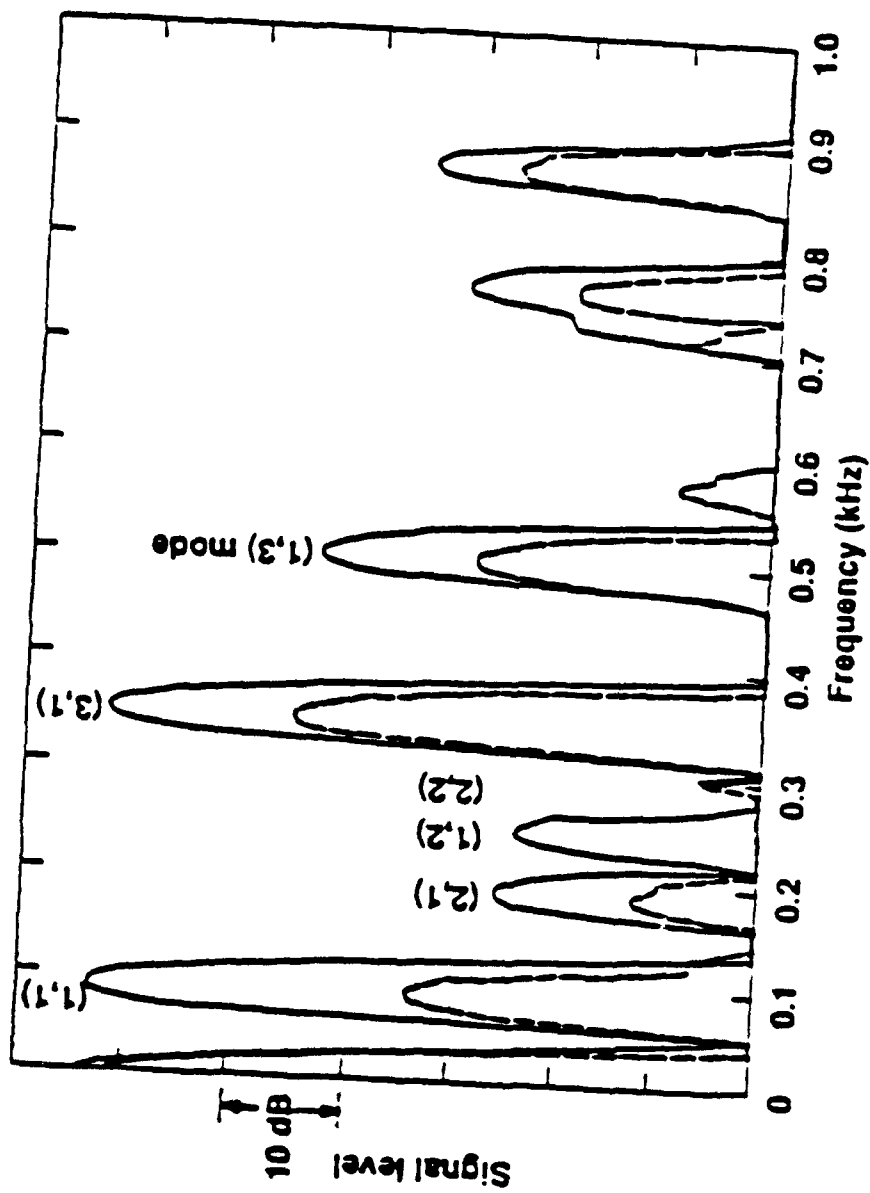


Figure 63. Piezoceramic sensing of panel response.
 —, piezoceramic; ---, accelerometer.

spectral content of the noise input. Other experiments on control of vibrational power flow in beams have also indicated such piezoelectric devices can be used to sense other control variables [64], such as vibrational response and power flow. Although the above devices can be successfully used to measure structural response, this data must be then manipulated to account for the structural acoustic coupling. A direct measurement of the radiated pressure field would be simpler and it is in this application that PVDF shows much potential. This material has been demonstrated to be of enough sensitivity to be able to measure acoustic pressure fluctuations and can be easily used as a distributed sensor where averaged effects (i.e., over a distributed control area) are required.

Piezoelectric polymers and co-polymers [65] such as PolyVinylidene Fluoride (2) (PVDF 2) thin films have gained popularity in recent years in the development of distributed actuators and sensors. Much interest has been given to the piezoelectric properties of these materials as the functionality and development of polymers in recent years has increased. They are compliant, lightweight, tough, plastic films that are available in a variety of sizes, shapes, and thicknesses, and configurations which includes bimorphs. It is highly sensitive over a large bandwidth (> 1 GHz), but it is noted for not being a highly efficient electromechanical transmitter due to its compliance.

Reference 19 discusses experimental work where PVDF distributed sensors were directly attached to vibrating panels and used as error sensors in an active structural acoustic control approach. When the PVDF sensors were shaped to only observe the odd-odd modes of the simply supported panel (i.e. the efficiently radiating modes), high global reductions in far-field sound were measured. This result should be contrasted to the use of point structural sensors such as accelerometers, which often lead to an increase in radiated sound levels [13]. In effect the shaped PVDF sensor acts as an analog structural wavenumber filter. If the sensor is long compared to the structure, then the PVDF sensor averages the response of high wavenumber, short wavelength, inputs (subsonic components) to zero while retaining information from the low wavenumber, long wavelength, inputs (supersonic components). As discussed previously a structural error sensor with these characteristics is highly desirable as the controller is only observing the critical radiating components of the motion of the structure. Associated with such approaches is a significant signal processing requirement in which the sensed data is converted into the required control variables.

6.3.1 MODAL SENSORS

VPI&SU has investigated under this grant, modal sensing of efficient acoustical radiators with PVDF distributed sensors in active structural acoustic control approaches [66]. The aim was to eliminate microphones in the far field by implementing PVDF thin film sensors on the surface of a plate. Since the far field acoustic radiation is dependent on the structural response of the plate, driving the response to zero would certainly result in zero acoustic radiation. In practice, this would be impossible, since an infinite number of control actuators would be required to couple with the plate response. Structures such as beams, plates, and shells have modes of vibration which do not couple effectively to the acoustic media. As a result these modes do

not efficiently radiate sound and thus do not require control. For comparison, the radiation efficiency for several modes of a simply supported plate are presented graphically in Figure 64.

The radiation efficiency approaches unity for all modes when the structural wavenumber is greater than the acoustic wavenumber in the surrounding media. When the structural wavenumber is greater than the acoustic wavenumber, the radiation efficiencies for the odd-odd modes (x and y) are the highest. However, the radiation efficiency of the even-even modes are the lowest, which is of interest in control of sound in this regime, since these modes will not contribute significantly to the radiated field. Only those modes possessing a high radiation efficiency must be observed and attenuated to effectively reduce the far field sound pressure level. In fact in some applications, particularly off-resonance, the response of the panel increases while sound radiation decreases. As a result of this observation, the dimensionality of the controller can be greatly reduced.

A control experiment was performed using multiple piezoceramic actuators and PVDF sensors bonded to a plate inside VPI&SU's anechoic chamber. The plate used in this experiment was configured with three sets of piezoelectric actuators as illustrated in Figure 65, and the control approach was based upon the filtered- x version of the adaptive LMS algorithm implemented in a TMS320C25 DSP processor for up to three channels of narrowband control. The actuator configuration allows coupling with odd modes for both the x and y directions, and each actuator consists of two piezoelectric elements of dimensions $38.1 \times 21 \times 0.19$ mm bonded symmetrically (front and back). The shape and location of the PVDF sensors are chosen such that the even-even modes are not observed. A signal generator was used to drive a shaker as the disturbance source and to provide the reference signal to the controller. The interconnections with the adaptive controller are illustrated in Figure 66. Experiments were performed for both on- and off-resonance cases of the plates, and some of these results will be discussed.

For the first case, the plate was driven on the resonance for the $(1,1)$ mode which is an efficient radiator, and both the acoustic and modal responses are shown in Figure 67. The acoustic far field was attenuated between 20 and 40 dB ("C#" indicates actuator used, and "PVDF#" indicates error sensor used). Considering the modally decomposed structural response, also depicted in Figure 67, the $(1,1)$ mode of the plate is obviously dominant before control. After applying control, a "modal restructuring" occurred (i.e., modes phased for constructive interference in the far field). For this case, the response of the $(2,1)$ mode was dominant under control conditions, and with the current sensor configuration it is weakly observed, and it is a poor acoustic radiator. In addition, the directivity of the acoustic response under control conditions can be attributed to the phasing between modes present in the structure. As a second case, the plate was driven on-resonance for the $(3,1)$ mode. After applying control, the acoustic response was attenuated by approximately 30 dB as can be seen in Figure 68. A driving frequency of 320 Hz was chosen for the off-resonance case, which lies between the $(1,2)$ and $(2,2)$ modes of the structure. Results are presented in Figure 69, where it is observed, approximately 10 dB of global sound attenuation was obtained after minimizing the response of the PVDF sensors. The modal response of the plate was observed to increase at every mode under control conditions, while the acoustic response was reduced indicating "modal restructuring". With the exception of the $(1,1)$ mode, the dominant response under control

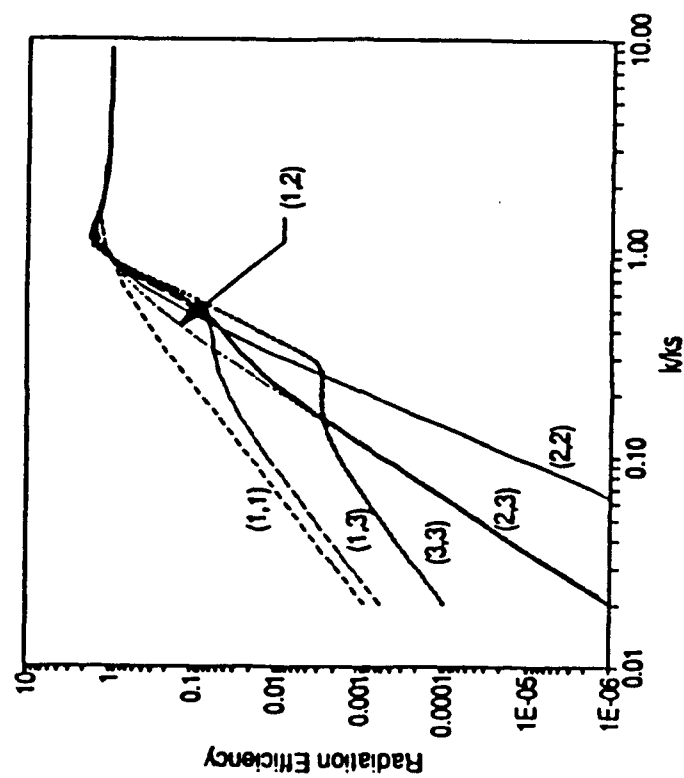


Figure 64. Radiation efficiency of plate.

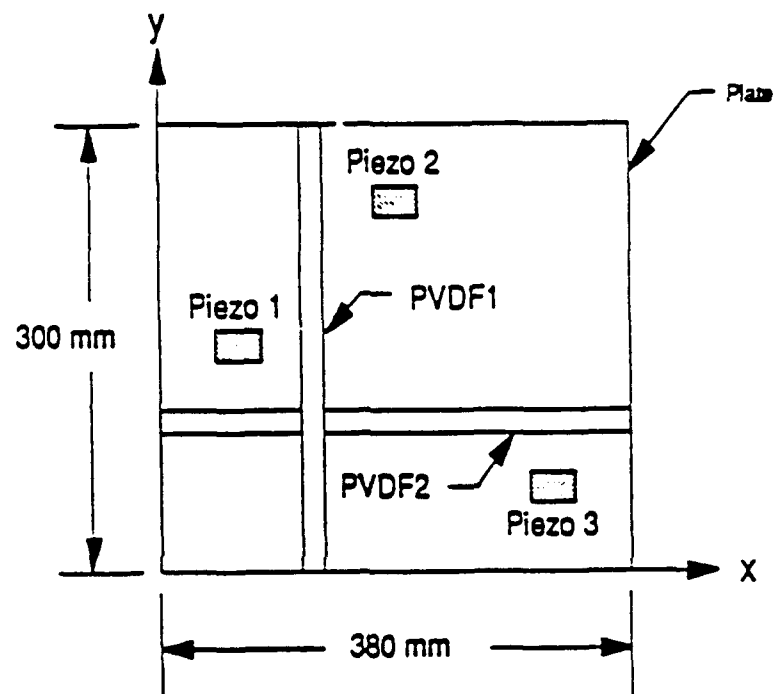


Figure 65. Schematic of plate.

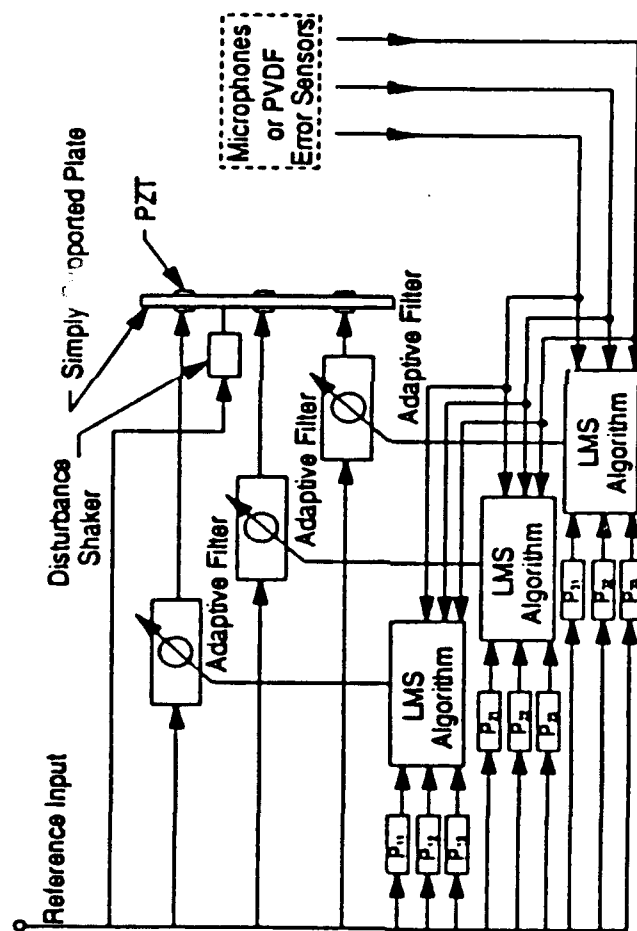


Figure 66. Block diagram of controller.

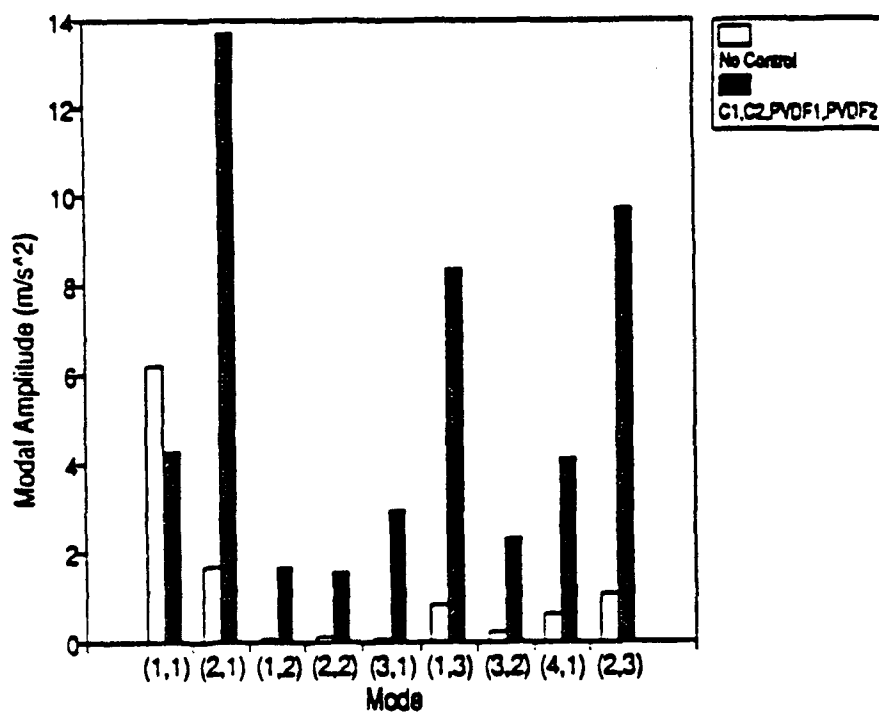
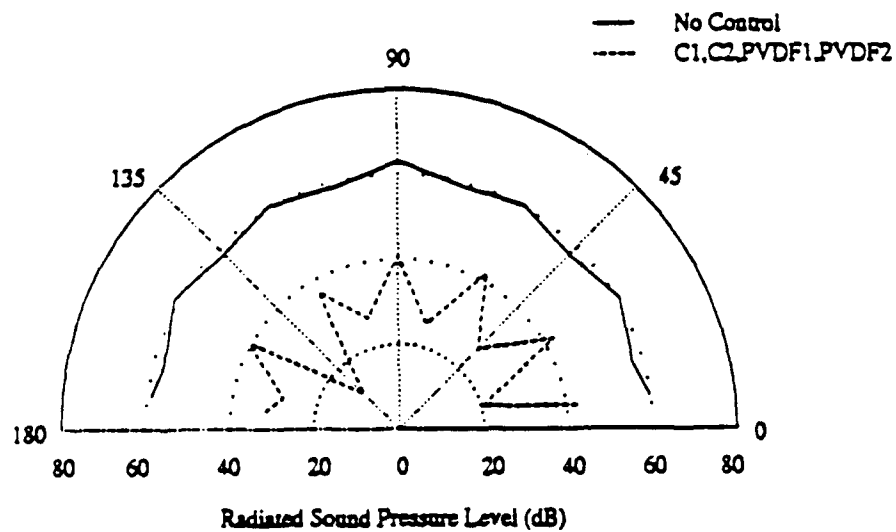


Figure 67. Response at (1, 1) mode (PVDF error sensors).

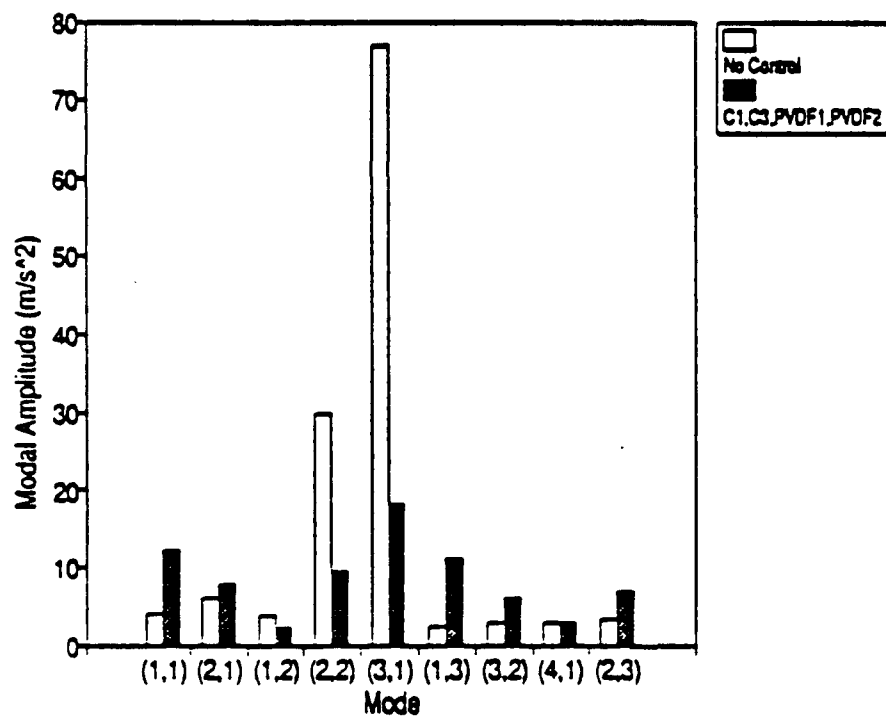
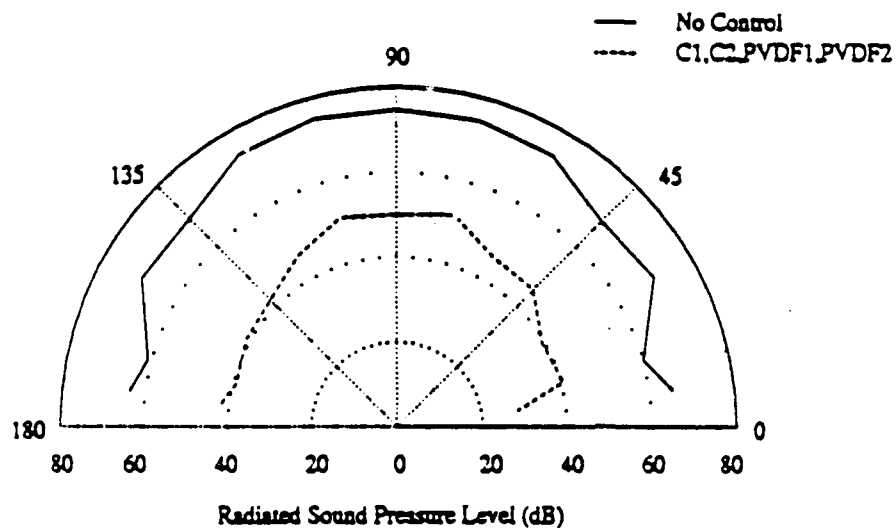


Figure 68. Response at (3,1) mode (PVDF error sensors).

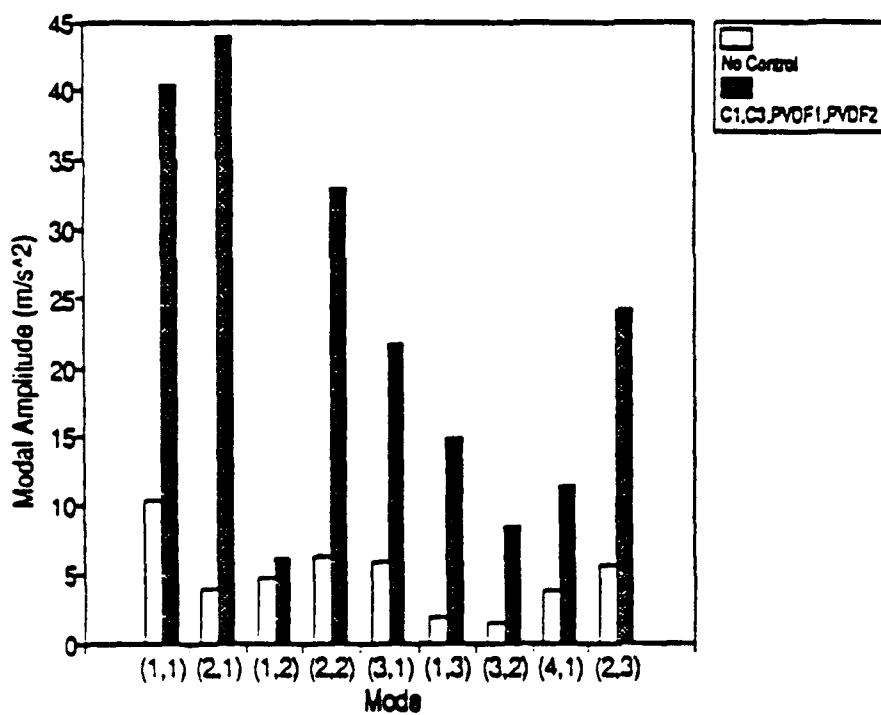
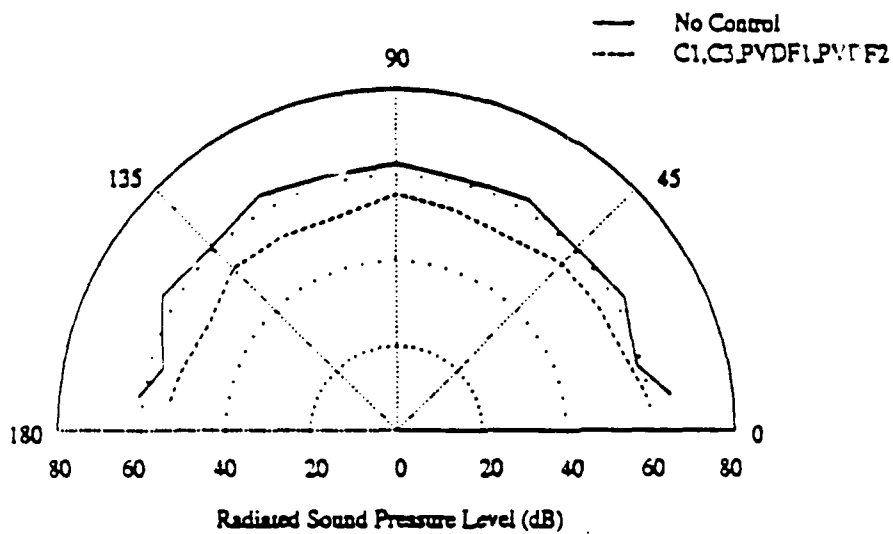


Figure 69. Response at 320 Hz (PVDF error sensors).

conditions was observed in the (2,1) and (2,2) modes of the structure, which are less efficient acoustic radiators and are poorly observed by the PVDF sensors. This spillover into the even modes is due to the chosen sensor configuration, since each sensor was located such that the even mode response in both the x and y directions would be rejected.

For comparison, Figure 70 shows the results using three error microphones in the acoustic far field when the structure was driven on-resonance for the (3,1) mode. The resulting acoustic response after control was observed to decrease by approximately 30 dB, as was the case when two PVDF sensors were used. This demonstrates a key advantage of PVDF structural sensing in reducing the number of control channels.

While the original goals of this Grant were to pursue active structural acoustic control of plates, it advanced well beyond that to the design of both actuators and sensors for cylindrical structures. This includes theoretical investigations and experimental results for active control of vibration and acoustic radiation. Many of the basic actuator and sensor developments mentioned previously have been translated to cylindrical structures, and this work is continuing with increased sizes in structures and experiment implementation.

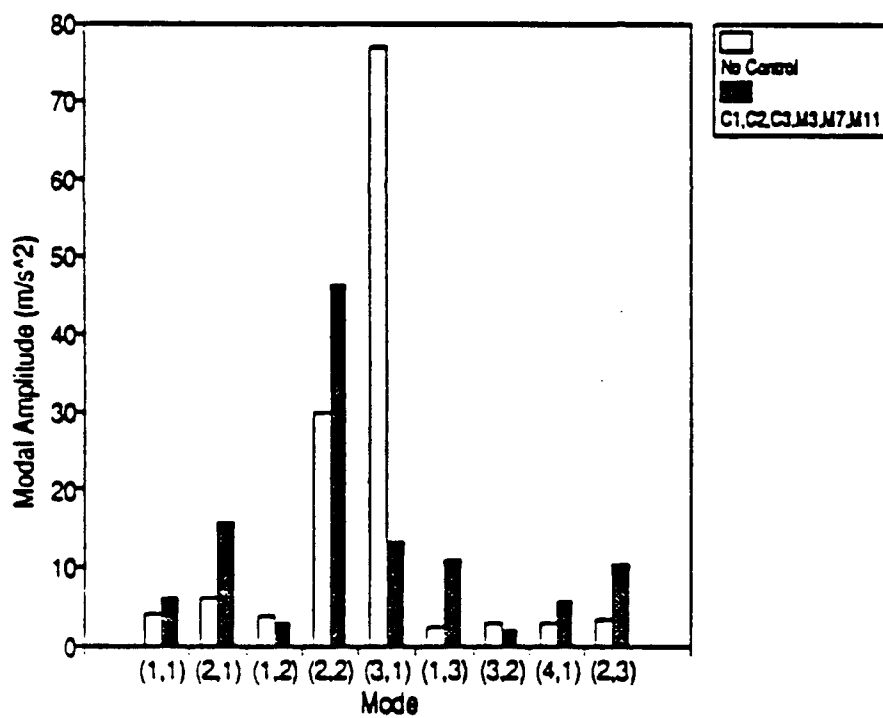
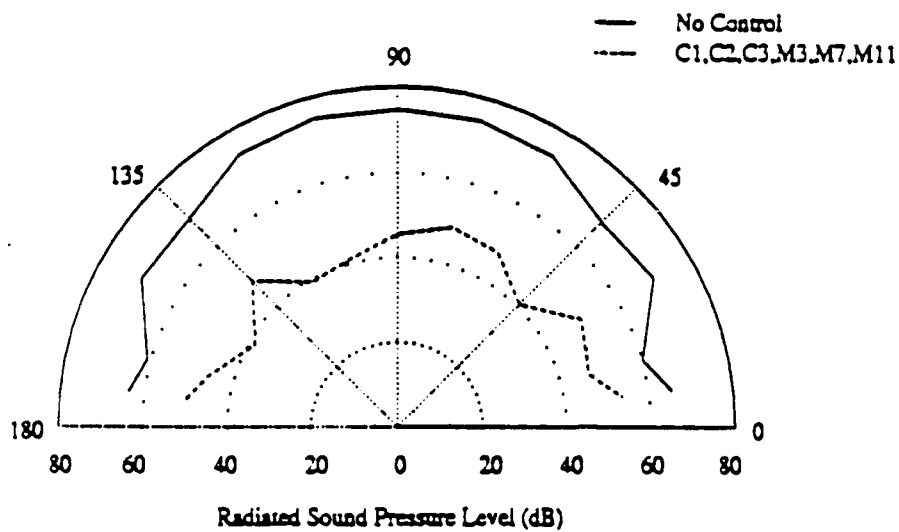


Figure 70. Response at (3, 1) mode (microphone error sensors).

SECTION 7.0

ACTIVE CONTROL TECHNIQUES

An important aspect of active control of sound radiation from structures is the choice, design and implementation of a suitable control strategy. Ultimate choice of strategy is dependent upon a number of factors but perhaps the most important is the nature of the noise input; whether it is steady state sinusoidal (including multiple frequencies), random or transient. All of these type noise conditions will be encountered in structural acoustic applications.

Terms such as active control and cancellation of acoustic noise can be easily interchanged and/or misused. To help clarify the meaning of the various active approaches some definitions are in order. "Control" is to optimally attain and maintain a desired (required) state (steady and/or dynamic). "Cancellation" is to null out a state by the superposition of an opposite polarity. Control has the broadest definition, while cancellation is more specific, and should be considered a subset of active control. "Control" does not necessarily mean that the acoustic field is reduced in level, but rather it is in a desired form. One example is the control of room or theater acoustic properties such as the reverberation time and spectral emphasis. However, most frequently it implies a minimization through control techniques which could be passive, semi-active, or fully active.

VPI&SU has developed control approaches which could be classified as semi-active and fully active techniques for the control of radiated noise. Passive techniques are defined by dissipation based upon material and structural properties, dependent upon only local properties, and can only dissipate and temporarily store energy. However, they are inherently stable, and do not require sensors, actuators, or controllers. Semi-active techniques on the other hand attempt to control with actuators the dissipation of energy using realtime or near realtime control with remote sensing. The semi-active control system response can be designed to be tailored to particular disturbance characteristics, and to adapt to changing conditions. Fully active techniques can additionally supply energy as well as dissipate it, and can be designed to be fully coherent with the disturbance, and remotely sensed information.

7.1 STATE FEEDBACK CONTROL TECHNIQUES

Research into the application of state feedback methods has focused on casting the structural acoustics control problem into the paradigms of modern control theory. In this research Linear Quadratic (LQ) techniques such as LQG, and LQR have been investigated theoretically and experimentally. This has included the development of Kalman filter state estimators. More recently techniques such as H_∞ have been addressed. Continuously persistent narrowband and wideband disturbances have been investigated as well as transients with considerable success experimentally.

7.1.1 LQR CONTROL

In one investigation [67], the LQR (Linear Quadratic Regulator) optimal method has been applied to the control of radiation from a subsonic baffled, clamped-clamped beam in air. The approach was to consider the transient radiation suppression of the structure when it has been excited by impulsive forces or initial conditions. The structure was described via an Euler-Bernoulli model of a uniform clamped-clamped beam. This PDE description was then discretized using a Ritz-type expansion in terms of the first three controlled modes of the beam.

The fluid medium, air in this case, was modeled by assuming a harmonic motion input from the beam structure as one argument of a Rayleigh integral written to yield the spatial, far-field pressure distribution. The resulting integral was then treated as a transfer function between the terms of the assumed beam spatial expression (the uncontrolled modes) and the far-field pressure distribution. This study treated only the diagonal terms (the direct terms) of the resulting spectral factorizations of the radiation resistance of the modes. Later work has introduced the off-diagonal terms which are

important in the presence of heavy fluid loading. These transfer functions were then implemented as filters in a real-time control analysis. Causality was ensured by using the inverse Laplace Transform to derive the filter impulse response functions. This formulation allowed the acoustic radiation to be inferred from the structural motions.

These relations were used as partial state estimators in a LQR optimal control formulation. The performance of the controller designed to reduce radiation (acoustic controller) was compared to the performance of a controller designed to reduce beam vibration. Equal energy was assumed for each case in order to normalize the comparisons. Figure 71 shows the modal time history of the beam with two force actuators used to suppress vibration. Figure 72 shows the result of the acoustic control; the radiation was reduced by 73% as compared to that of the vibration controller. Increased attenuation could be achieved by decreasing the penalty factor on control effort. Note that the acoustic controller allows mode two, an inefficient radiator, to ring much longer than was allowed by the vibration controller; it, instead, puts its energy in to the more efficiently radiating third mode. This work is being extended to treatment of the radiation coupling of the modes and to experimental realizations.

7.1.2 SMA COMPOSITE BEAM CONTROL

Other work with state feedback control of structural acoustic radiation involves the control of SMA-reinforced composite beams [53]. Part of this experimental work, the pattern search minimization, was discussed earlier in section 5.2 of this report. Another portion of this work was concerned with the closed-loop tuning of the beam radiation response anywhere within an octave bandwidth above the fundamental mode. It has been demonstrated [46] that the frequency of the lower order vibration modes of a clamped-clamped SMA composite beam are linearly related to the bulk temperature of the beam. The control approach was to use the measured temperature as a controlled variable to indirectly control the beam radiation. This experimental

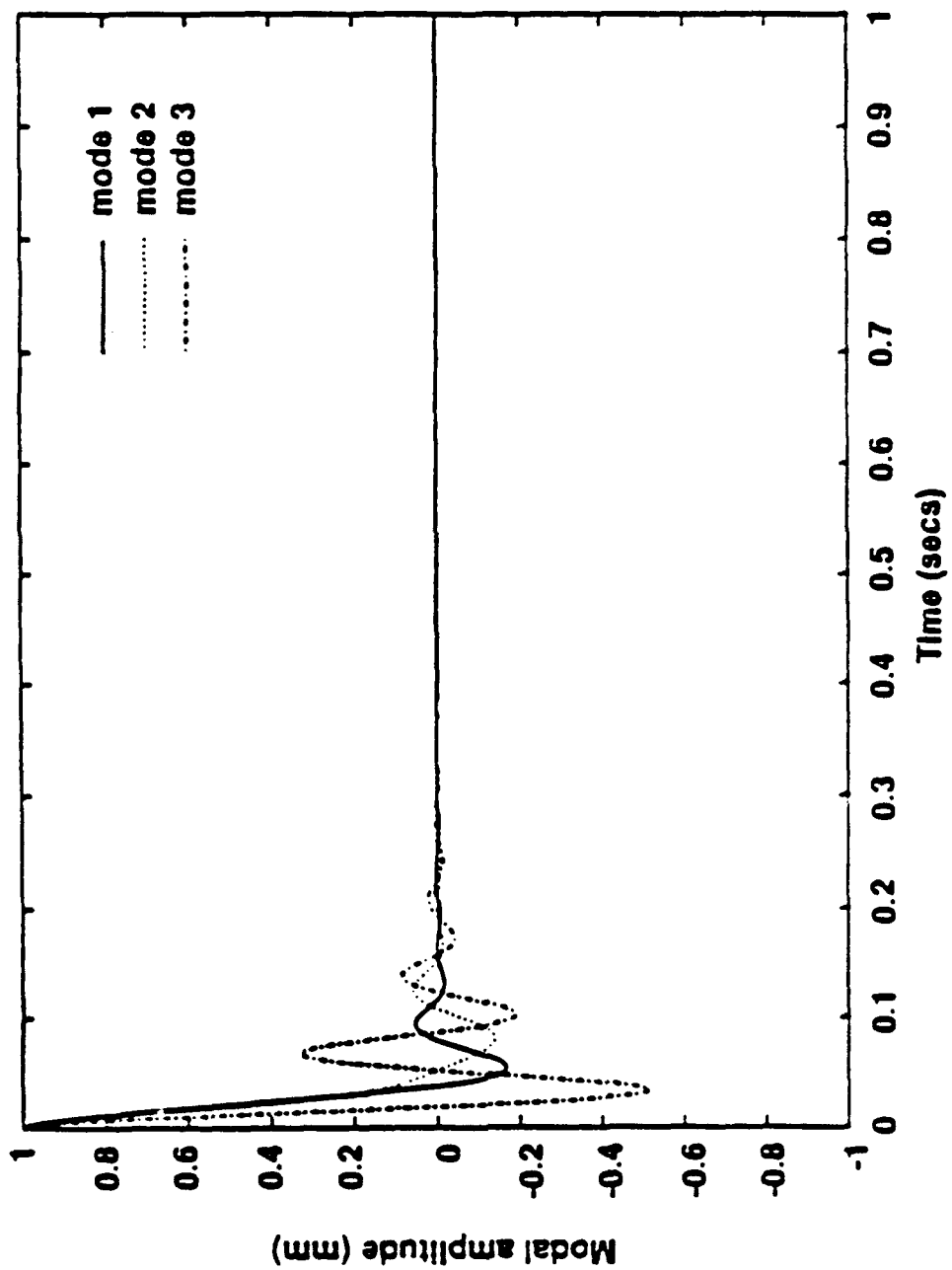


Figure 71. LQR control of beam response.

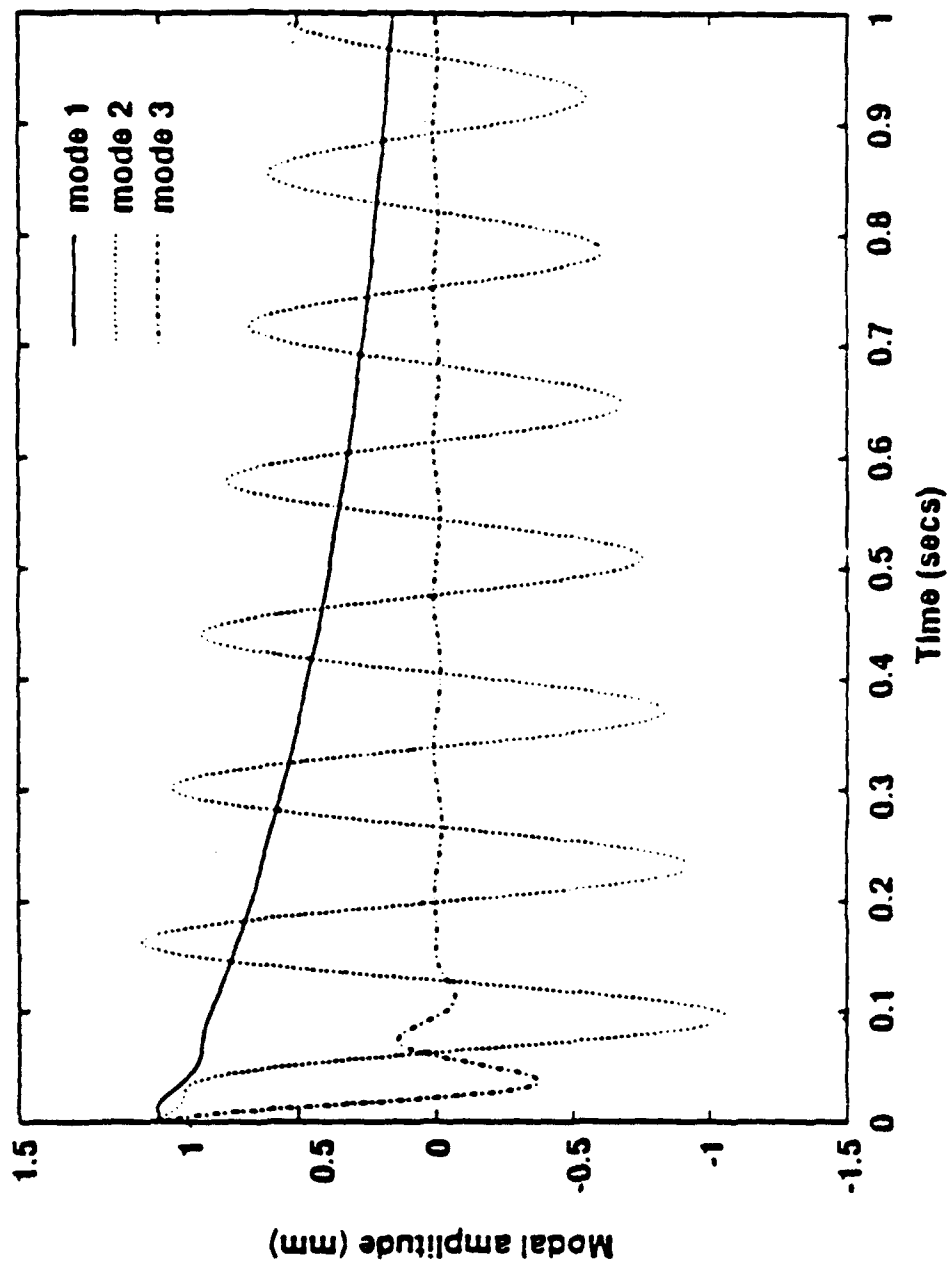


Figure 72. LQR control of beam sound radiation.

and analytical study had as its goal to maximize the radiation response of the beam subjected to a steady-state disturbance.

The beam was modelled with a first-order, thermal capacitance model. The control was accomplished via a PC/AT machine which adjusted the amount of resistive heating of the SMA filaments in the composite structure to reach and hold a preset temperature. Figure 73 shows the results of the control with the beam being driven at 46 Hz, which is off the 35 Hz unactuated fundamental mode frequency. The closed-loop control increased the radiated sound pressure level by 20 dB by moving the fundamental frequency of the adaptive beam close to 46 Hz. This demonstrates the ability of the feedback control scheme to place the beam resonant frequencies at a setpoint.

7.1.3 TRANSPUTER CONTROLLER

VPI&SU has designed and implemented a transputer based digital controller to conduct state space multi-input multi-output (MIMO) realtime active structural acoustic control experiments [68]. The transputer is unique in that a number of transputer processing elements can be easily set up into a complex network for high speed parallel processing architectures. This has many advantages for advanced control system implementations and expansion into higher order and bandwidth control systems. This controller was implemented using a 386/387 style computer host equipped with a parallel processing transputer-based system to perform both control law calculations and the analog to digital conversion (A/D) and D/A operations. Figure 74 shows the overall experimental configuration used with the transputer based system. The transputer data acquisition board, designed and implemented by VPI&SU, has 16 channels of analog input and 16 channels of analog output with simultaneous high speed sampling and conversion. Empirical results have shown that good modal identification requires a sampling frequency approximately ten times faster than the vibration frequencies of interest. The multitasking ability of the control system architecture allowed the control law to be calculated on a single T800 transputer while the analog conversion operations were handled by a separate T222 transputer. This transputer system has been used in many of the MIMO feedback control experiments discussed in this report.

7.1.4 LQG CONTROL

The effectiveness of a low order vibration controller formulated using Linear Quadratic Gaussian (LQG) modern control theory has been demonstrated [68]. This is an important consideration given the amount of computational performance required for a high order controller. Simulator and experimental results were obtained for a two- mode compensator which had an eighth- order state-space model. A two-mode model was chosen to ensure that the spatial and temporal sampling requirements could be met by the implemented transputer based controller. The experiment was conducted with the controller using a cold-rolled steel plate, 0.6 m long, 0.5, and 0.003 m thick attached along its edges to emulate a simply-supported boundary condition. A disturbance input to the plate was applied as a point force using an electromagnetic vibration shaker. The response was measured using twelve accelerometers arranged in a nonsymmetric

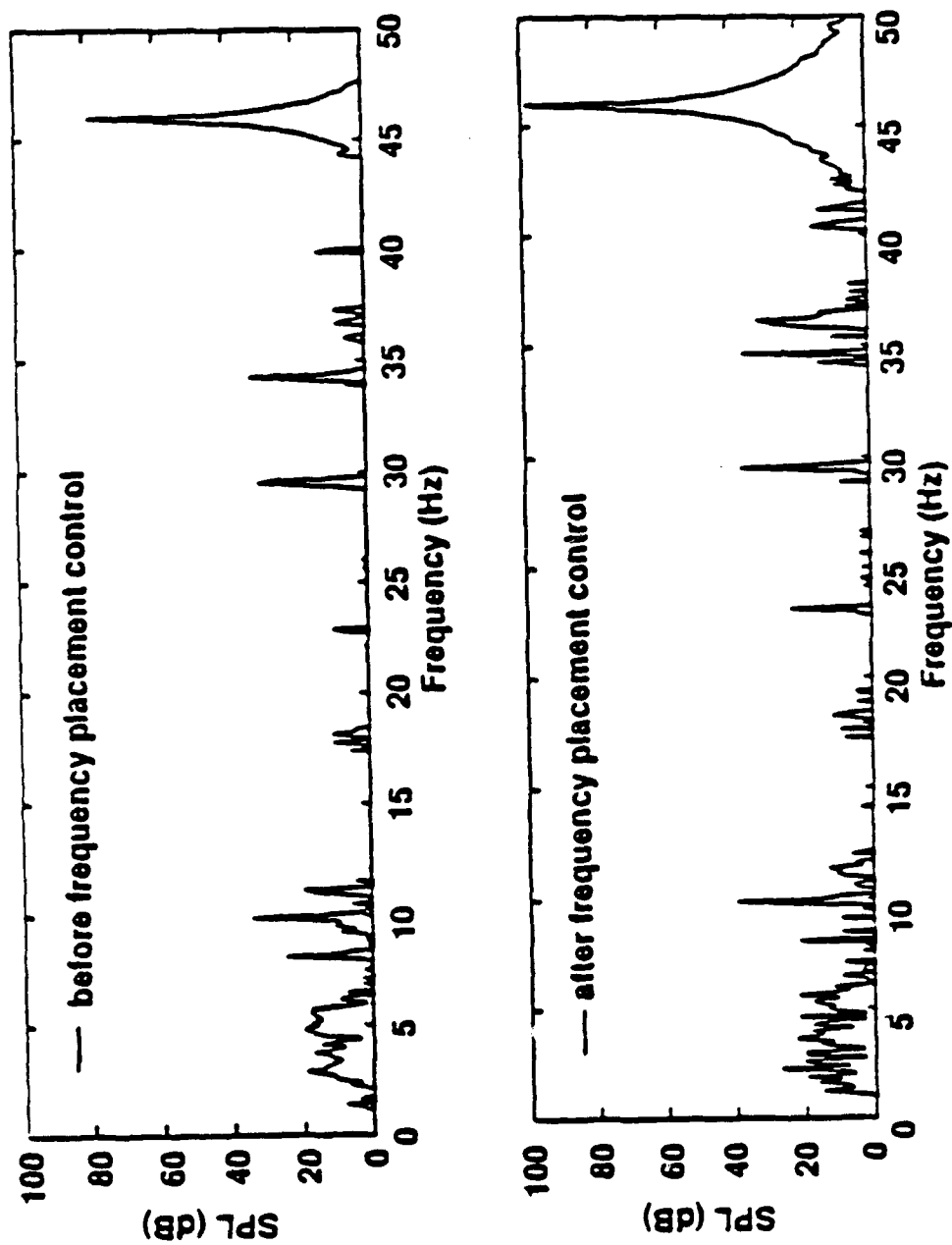


Figure 73. Peak radiation frequency placement.

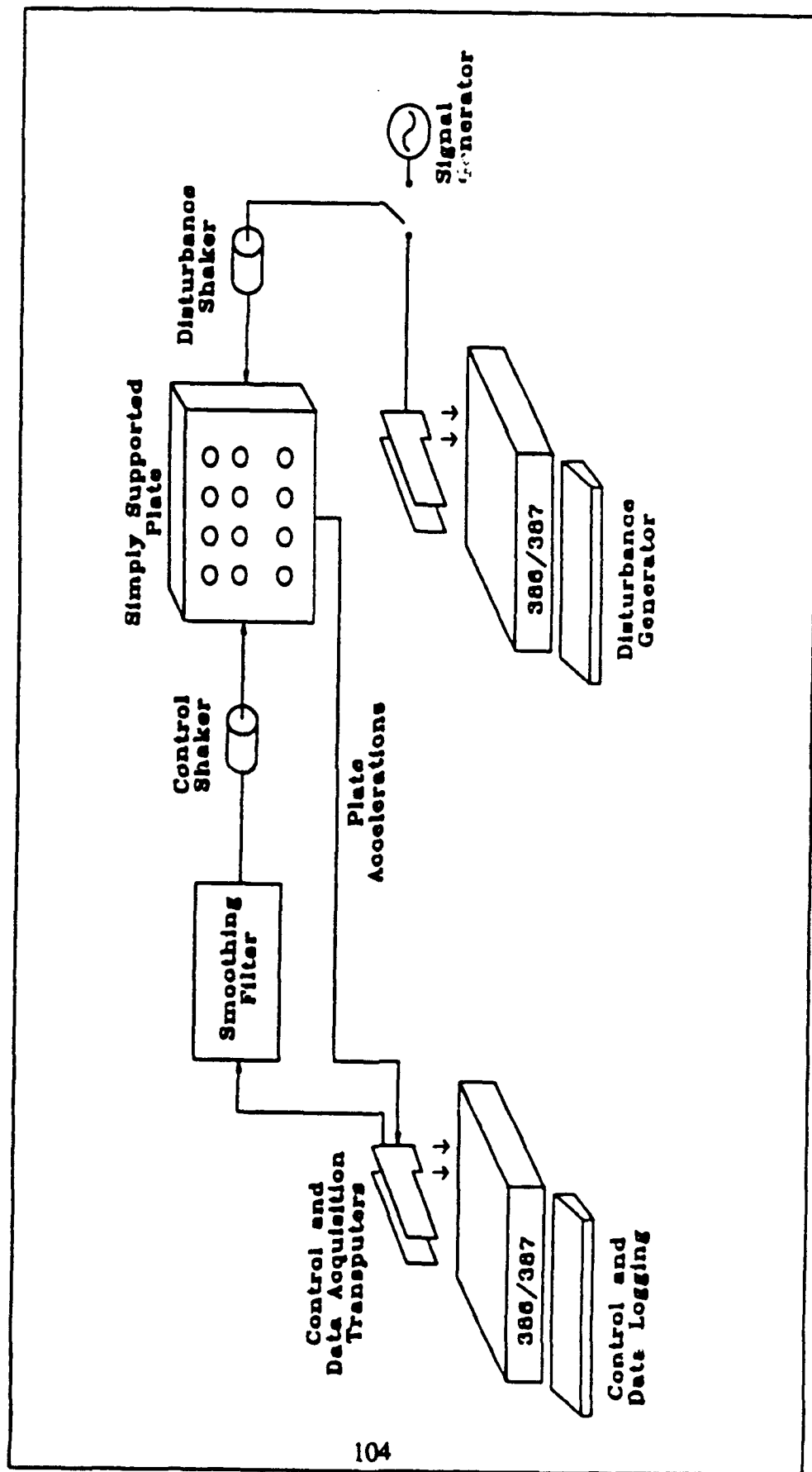


Figure 74. Experimental setup.

array to insure observability of the first twelve plate modes. Figure 75 shows the experimental time history results for a 60 Hz harmonic disturbance for Mode 1 and Mode 2, and Figure 76 shows the results for a narrowband disturbance centered at 60 Hz. Substantial disturbance rejection can be observed for the first mode amplitude showing that a low order controller can be designed without a degradation in performance. Transient suppression by the compensator was demonstrated by applying an impulse to the center of the plate during closed-loop control. A time history of the results are shown in Figure 77. Notice that the disturbance was minimized before the transient was applied to the plate. As the impulse was applied the, the mode 1 response damped out quickly, at a time scale corresponding to the closed-loop eigenvalue. The amount of additional damping introduced by the controller is obvious in the comparison plot in Figure 77. The compensator introduced approximately twice as much damping to the fundamental mode compared to the open-loop decay.

7.1.5 STATE-SPACE RADIATION FILTER

While the previous approaches considered primarily only vibration control, the modern control theory approach has been extended to apply to the control of acoustic radiation [69]. More specifically control of total radiated energy with the constraint that the farfield pressure cannot be directly measured, but rather estimated from measurements of the structure. Using this estimate as a cost function, a feedback controller is designed using Linear Quadratic Regulator (LQR) theory to minimize the cost. Computer simulations of a clamped-clamped beam show that there is an appreciable difference in the total radiated energy between the a controller designed to suppress vibrations of the structure and a system with a controller that takes into account the coupling of these vibrations to the surrounding fluid. The first step is the design of a "state-space radiation filter" which relates the structural states to the farfield acoustic field energy. To illustrate the idea an Euler-Bernoulli model for a uniform bar in a baffled, clamped-clamped configuration has been considered. To estimate the radiation, velocity measurements of the structural vibration are converted into velocities of the predetermined spatial functions and these velocities are filtered to produce an output signal whose integrated square is proportional to the radiation. For a comparison it is assuming the beam is excited by a pulse at a single point located at one fourth the overall length where a single actuator is also placed. The time histories of the modal amplitudes using a vibration controller and the acoustic controller are shown in Figures 78 and 79 respectively. The acoustic controller works harder to suppress mode 3 than does the vibration controller and leaves mode 2 virtually untouched. For the time interval considered the acoustic controller reduced the total energy radiated by 38% compared to the vibration controller. This matches our intuition based on the radiation efficiencies of the modes. The second simulation assumed that there were two actuators located at 0.29, and 0.5 the length with the disturbance pulse applied to the 0.29 location. The time histories of the modal amplitudes are shown in Figures 80 and 81. In this case the acoustic controller reduced the total acoustic energy by 69% compared to the vibration controller. Both controllers used more energy than in the first case, as can be seen by the increased damping of the modal amplitudes, but this resulted in an even greater advantage for the acoustic controller. The results of this work provide a framework for a general, model based method for actively suppressing transient

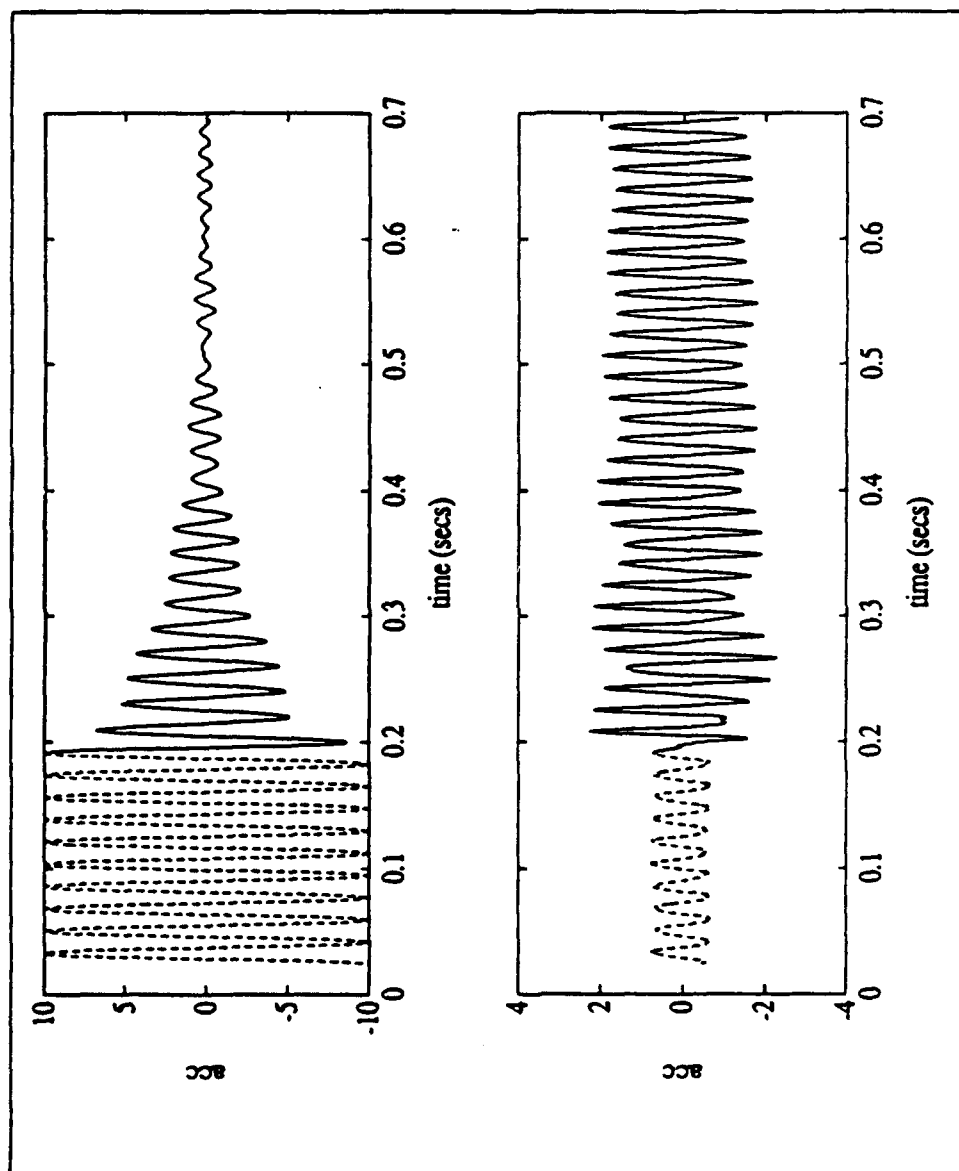


Figure 75. Experimental: 60 Hz disturbance compensation for mode 1 (upper) and mode 2 (lower).

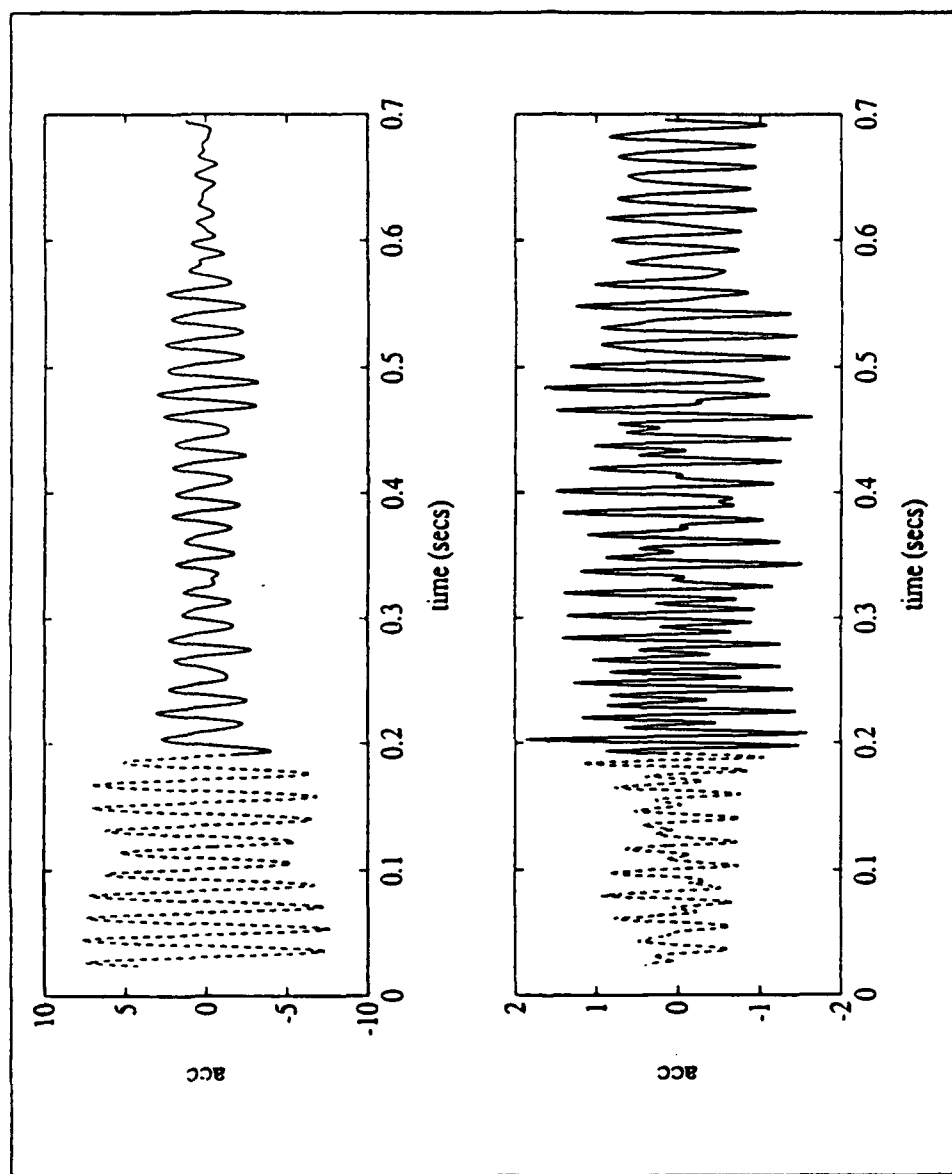
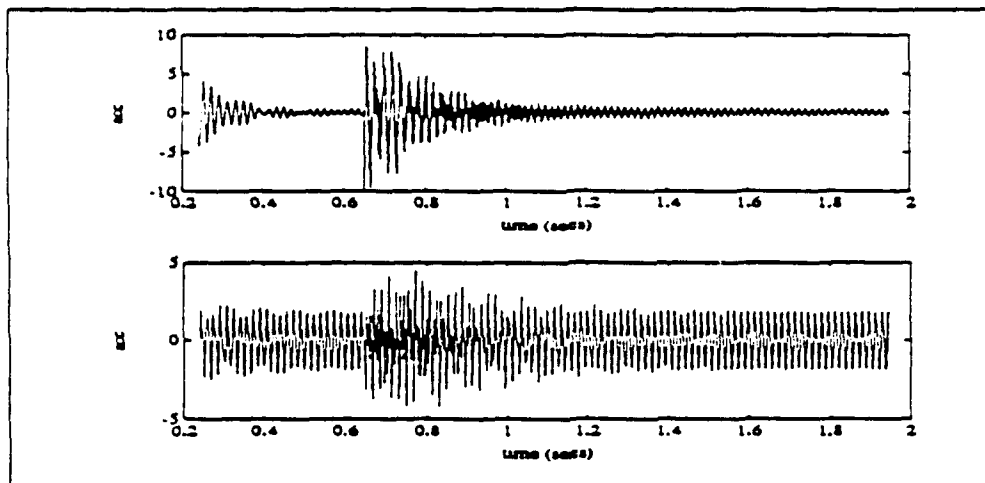
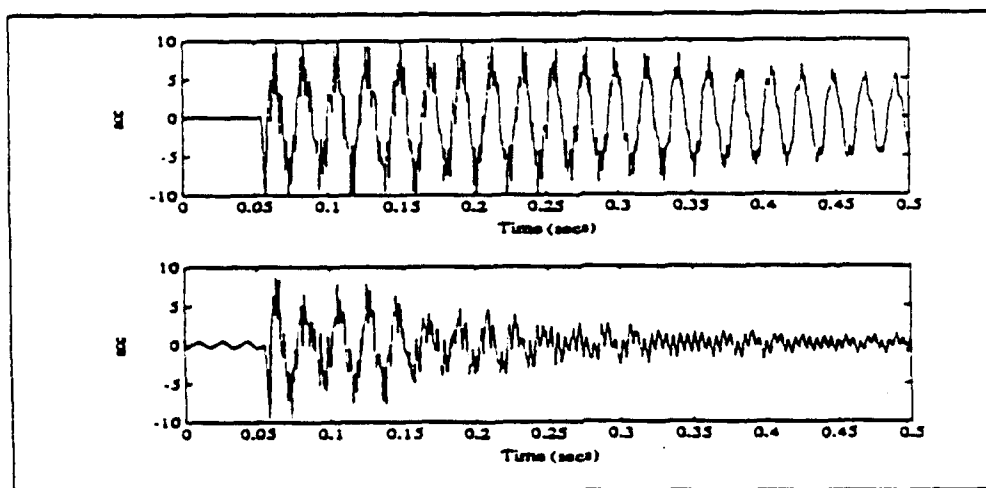


Figure 76. Experimental narrowband disturbance compensation for mode 1 (upper) and mode 2 (lower).



Mode 1 and Mode 2 Closed-Loop Transient Response



Mode 1 Transient Response, open-loop (upper) and closed-loop(lower)

Figure 77. Transient response.

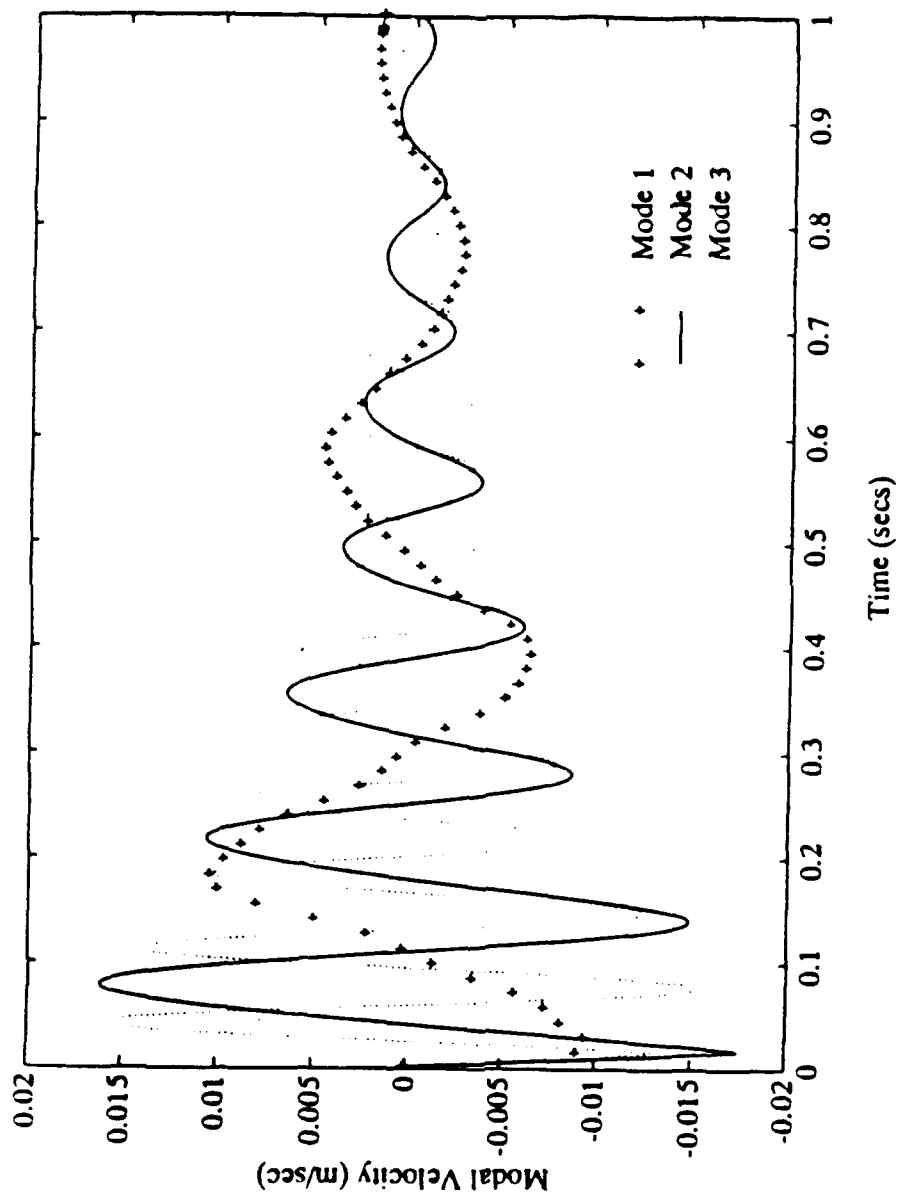


Figure 78. Modal time histories using vibration control, one actuator at $x = 0.25L$.

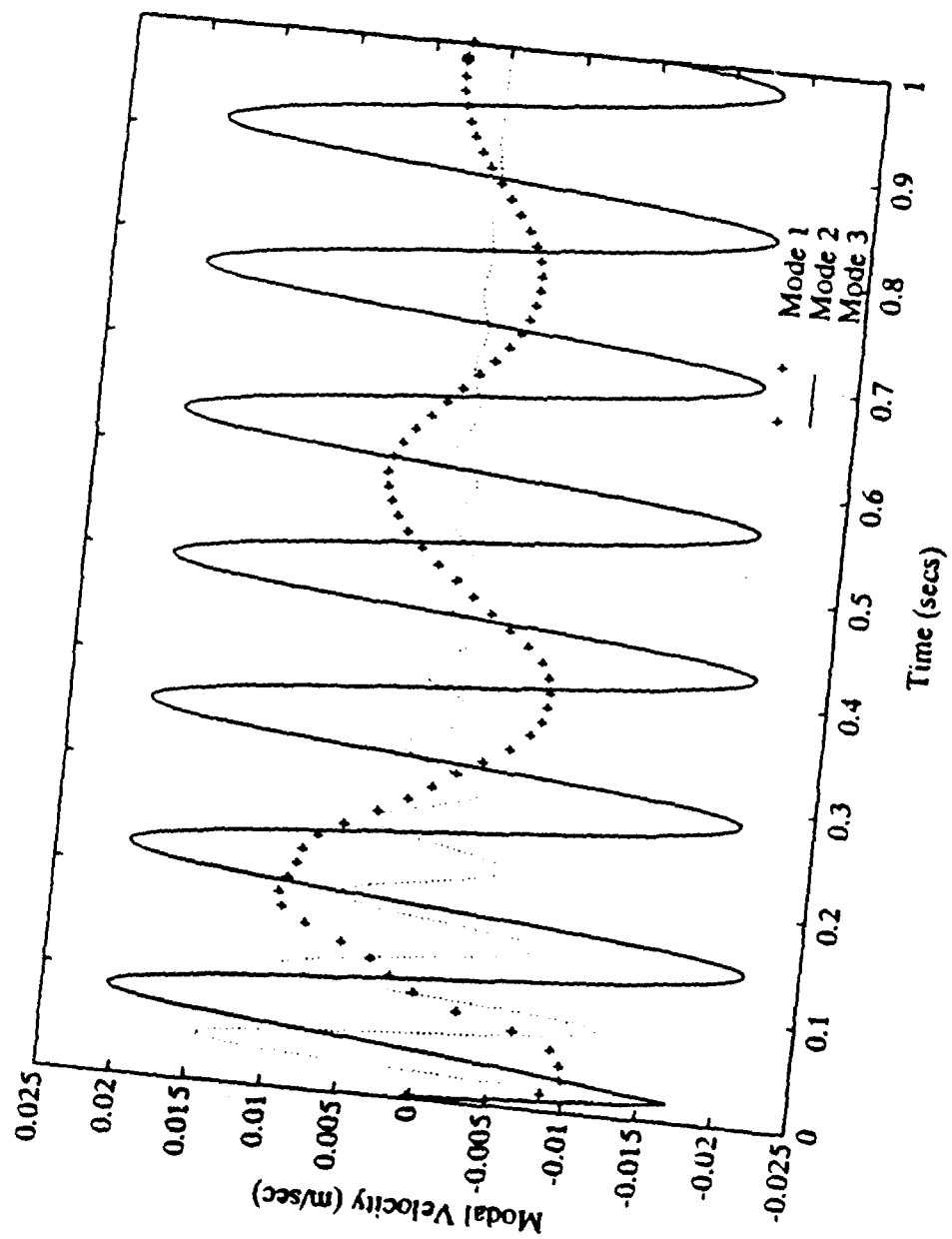


Figure 79. Modal time histories using acoustic control, one actuator at $x = 0.25L$.

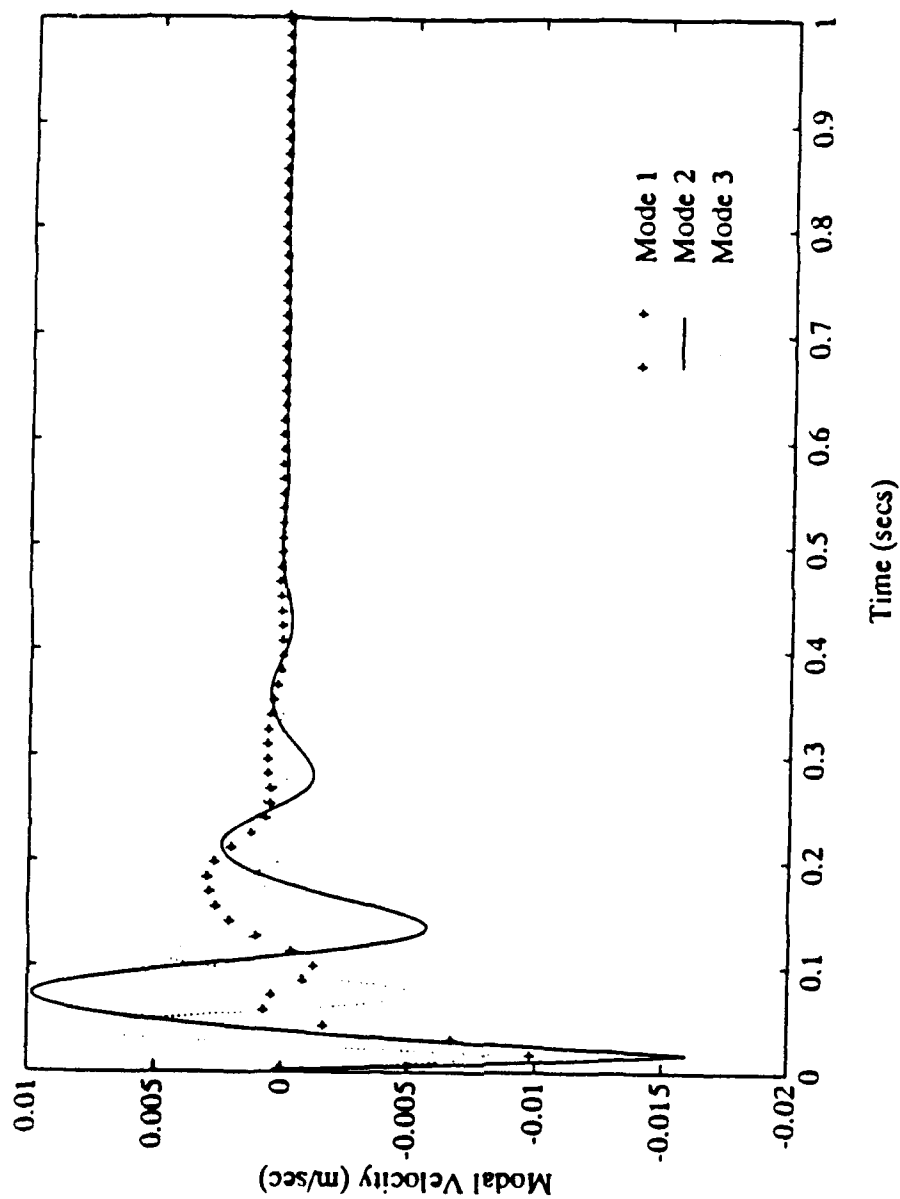


Figure 80. Modal time histories using vibration control, two actuators at $x = 0.29L$, $x = 0.5L$.

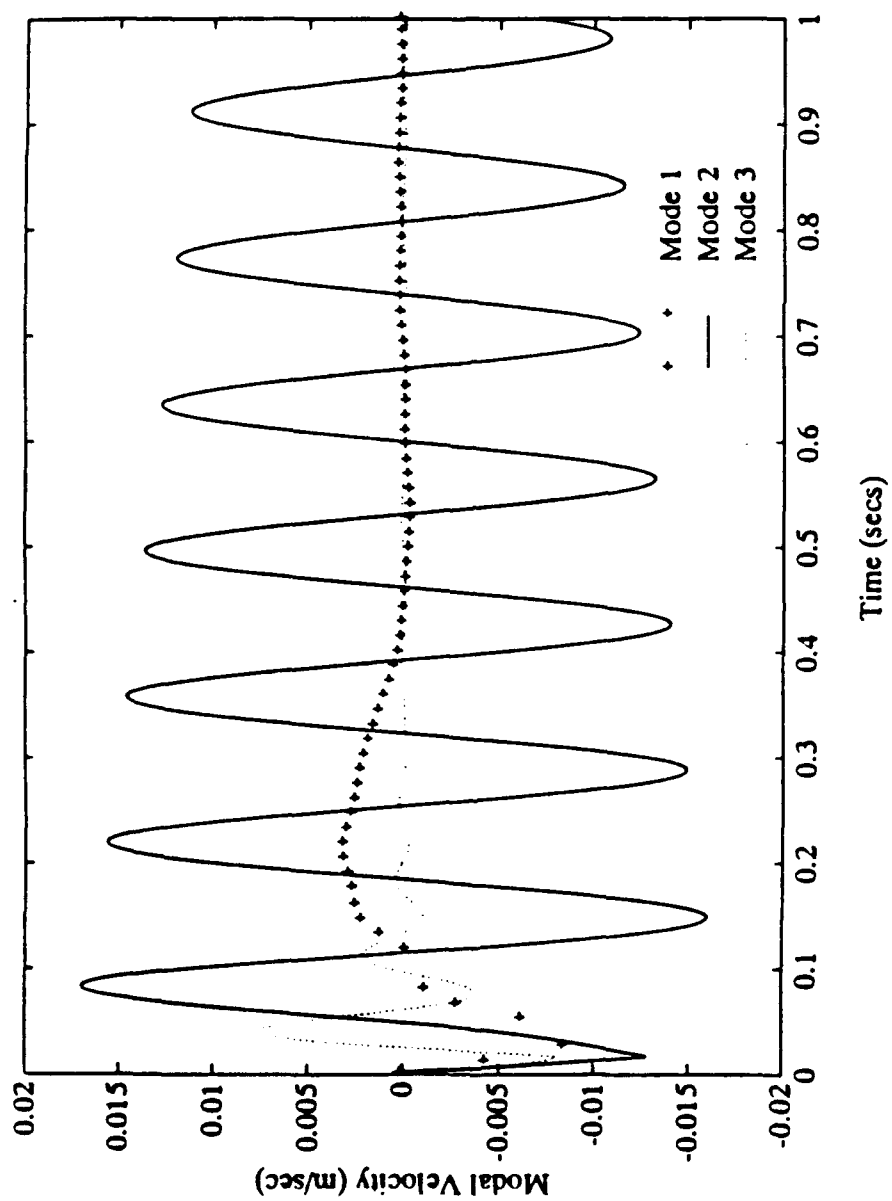


Figure 81. Modal time histories using acoustic control, two actuators at $x = 0.29L$, $x = 0.5L$.

structural acoustic radiation that can be also applied to steady, narrowband, or broadband disturbances.

7.2 LMS ADAPTIVE ALGORITHMS

For applications in which the noise field is a steady state sinusoidal input (or multiple frequencies) and in some cases random broad-band, the feedforward least mean squared (LMS) adaptive approach has proved quite successful [70]. Early active acoustic applications of the adaptive LMS algorithm were one-dimensional, and much of this work is summarized in the review article by Warnaka [71]. More recently it has been extended to multi-dimensional acoustic fields [72] as well as structural radiated noise [14]. In general, this approach relies on constructing a quadratic cost function by squaring the modules of the error variables and then using various techniques such as steepest descent, pattern search, etc., to find the unique minimum of the cost function. The control approach may be implemented in both the time or frequency domain. An advantage of the LMS approach is that unlike optimal control, little system identification is needed. An important consideration is that a good spectral estimate of the noise signal is needed. However, in many applications this can be either measured or computed directly from the noise input excitation (such as, for example, fundamental propeller frequency is directly related to shaft speed and number of blades).

Another important aspect of the feedforward LMS control approach is that, in general, as contrasted to the optimal approach, it relies on control inputs to the structure which may be viewed as having all the mass, spring, damper parameters of an attached sub-structure. Thus the controller can be viewed as performing "system modification" to lower the structural response by altering the system input impedance to the noise source. The modified input impedance thus generally results in lower noise energy transmitted into the control field. Recent analytical work has also demonstrated that, analogous to feedback controlled systems, the feedforward controlled system has new eigen-properties [73].

7.2.1 FILTERED-X ALGORITHM

Figure 82 shows the arrangement of typical time domain LMS adaptive approach based on the "filtered-x" algorithm [70]. The heart of the system is the adaptive FIR filter whose coefficients are updated by the control algorithm in order to minimize the signals at the error microphones. In this case the arrangement corresponds to the experiments of Ref. 13 with corresponding results of Figures 5 and 6. The data acquisition and control algorithm in this application were written in assembly language and down loaded into a TMS320C25 chip in conjunction with three A/D converters and two D/A converters for dedicated control implementation. The number of coefficients in each filter was two, thus limiting the application to narrowband. The sample rate was fixed at 2 kHz. This arrangement enabled flexible reprogramming of the control approach as well as high convergence speed.

Figure 83 shows a typical time history of the error signals and the system can be seen to converge in approximately 50 msec to approximately 15 dB of attenuation when control is

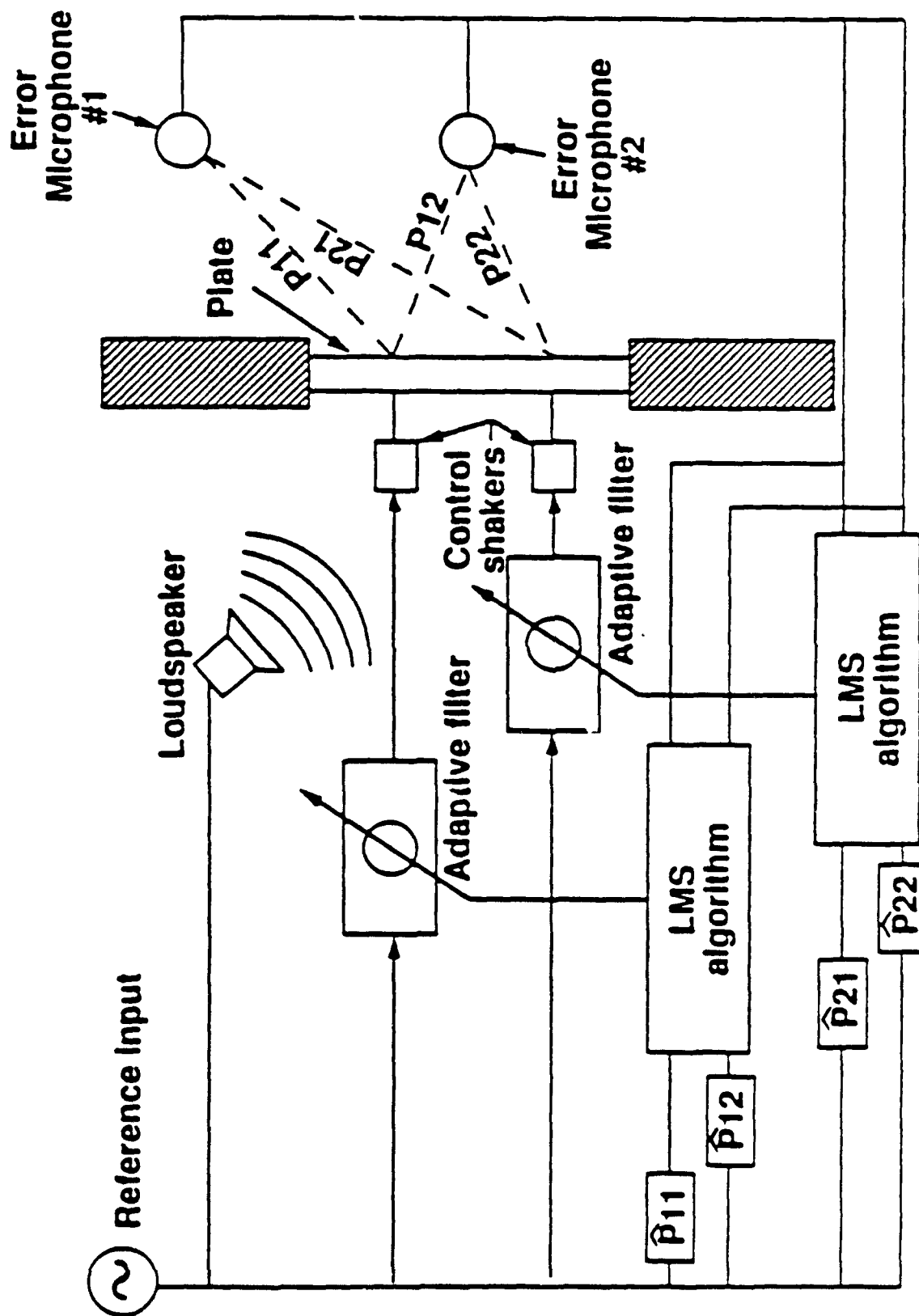


Figure 82. Arrangement of time domain adaptive LMS controller.

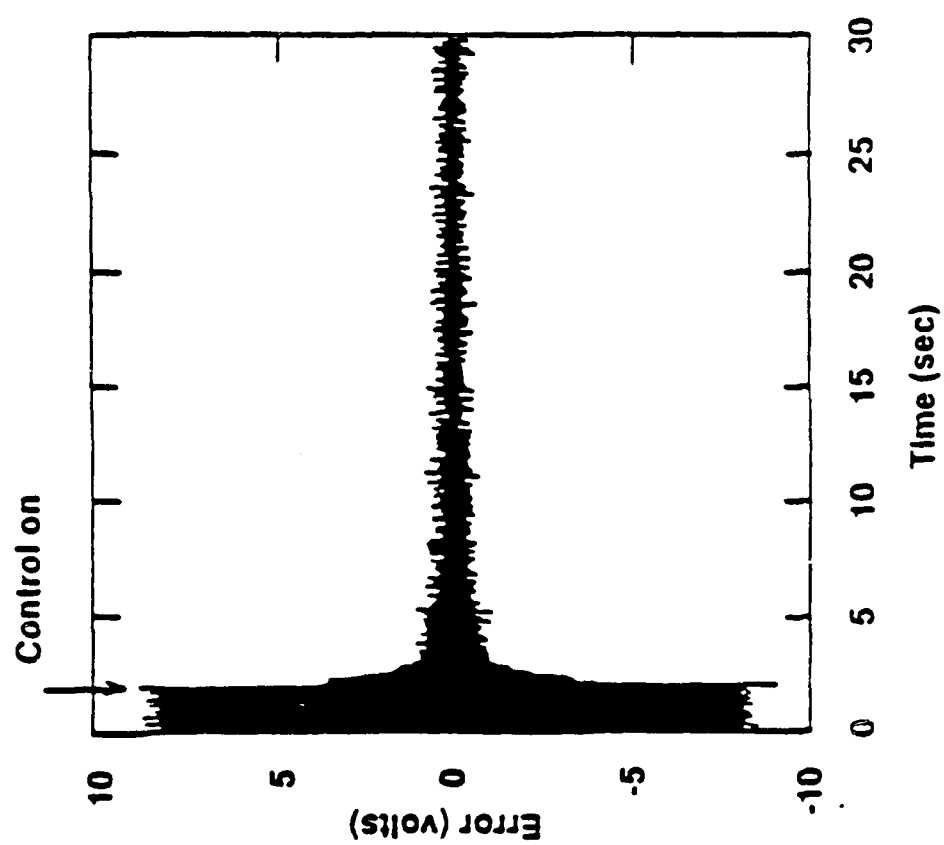


Figure 83. Error signal of adaptive LMS controller.

switched on. Thus such controllers can effectively adapt and track many structural acoustic inputs (e.g. an aircraft engine) in "real time". Other LMS adaptive approaches rely on constructing a cost function from frequency domain information. In this case it is not necessary to estimate the phase delay between the control inputs and error sensors as this is averaged and as many error sensors as required can be easily used. However, this approach is generally slower than the time domain due to the higher sampling requirements. Figure 84 shows a typical control variable path for a frequency domain LMS adaptive controller as the system searches for the quadratic cost function minimum [74]. This system was used for control of structure-borne sound in aircraft [75] and while it is slower to converge than the time domain approach it is easier to implement and more stable with respect to use of multiple sensors.

7.2.2 BROADBAND FEEDFORWARD

Two adaptive feedforward control structures based on the filtered-x LMS algorithm have been developed for active control of broadband vibration in structures [76]. Figure 85 illustrates the conventional Filtered-X Control Configuration where the adaptive process is stabilized by filtering the reference signal (X_p) by an estimate of the control input (U_p) to error (e_p) transfer function (T_{ω}). In the first control structure considered, the conventional Filtered-X Control Configuration is modified such that the transfer function between the control input and the error output is represented by an infinite impulse response (IIR) filter $[A(z)/(1-B(z))]$. An IIR filter will most efficiently model the resonances and antiresonances which are characteristics of a structure, but they introduce stability requirements that must be satisfied (i.e., the poles must be inside the unit circle). In order to remove these stability problems, the Equation Error Control Configuration is developed by first filtering the error signal by the poles of the system, $[1-B(z)]$, before minimizing it as shown in Figure 86. The system identification process for determining the poles and zeros of the plant and subsequent filters required is done using autoregressive-moving average (ARMA) models. The control signal is obtained in both configurations by filtering the reference signal through an adaptive finite impulse filter (FIR).

Both control configurations were experimentally investigated on a simply-supported beam and power reductions of up to 20 dB were observed. Figure 87 shows the simply-supported beam with sensor and actuator locations, and Figure 88 shows the overall setup of the experimental equipment. Control and disturbance inputs were applied using G1195 piezoelectric strips. It was observed that the controller produced better results when velocity rather than acceleration was used as the error signal. The two control approaches were implemented in a Texas Instruments TMS320C30 digital signal processor (DSP) board installed into a host 80386-based personal computer (PC). A programmable delay z^{-d} was installed in the disturbance path allowing the control system to be casual by varying the delay d to give at least 3 ms of delay to account for the measured delay difference between the control and disturbance at the error sensor.

The experiments were limited to control of the first three bending modes because the location of the control actuator pair rendered the fourth mode uncontrollable, and thus the frequencies

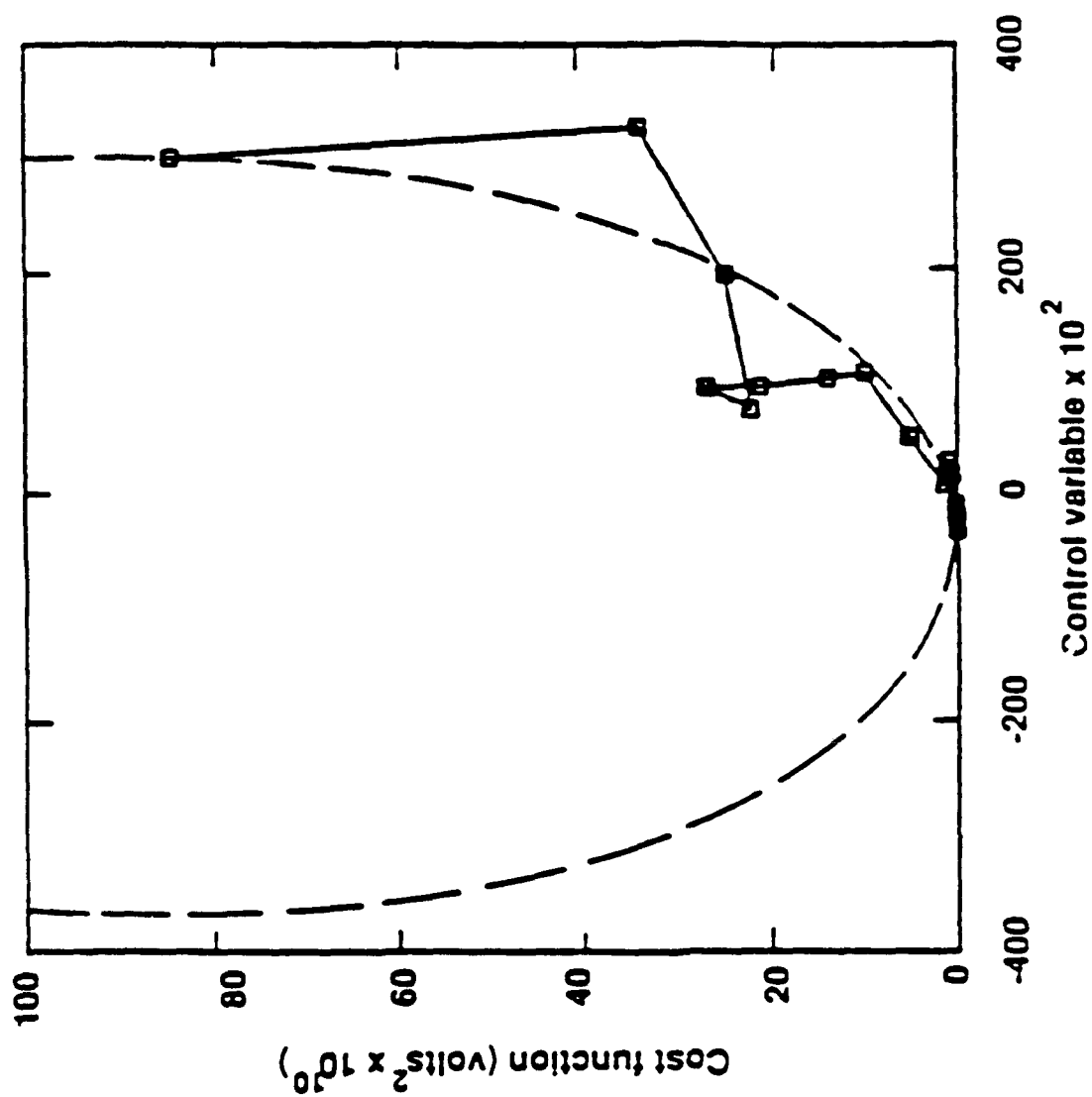


Figure 84. Adaptive controller characteristics. —, cost function; —, controller path.

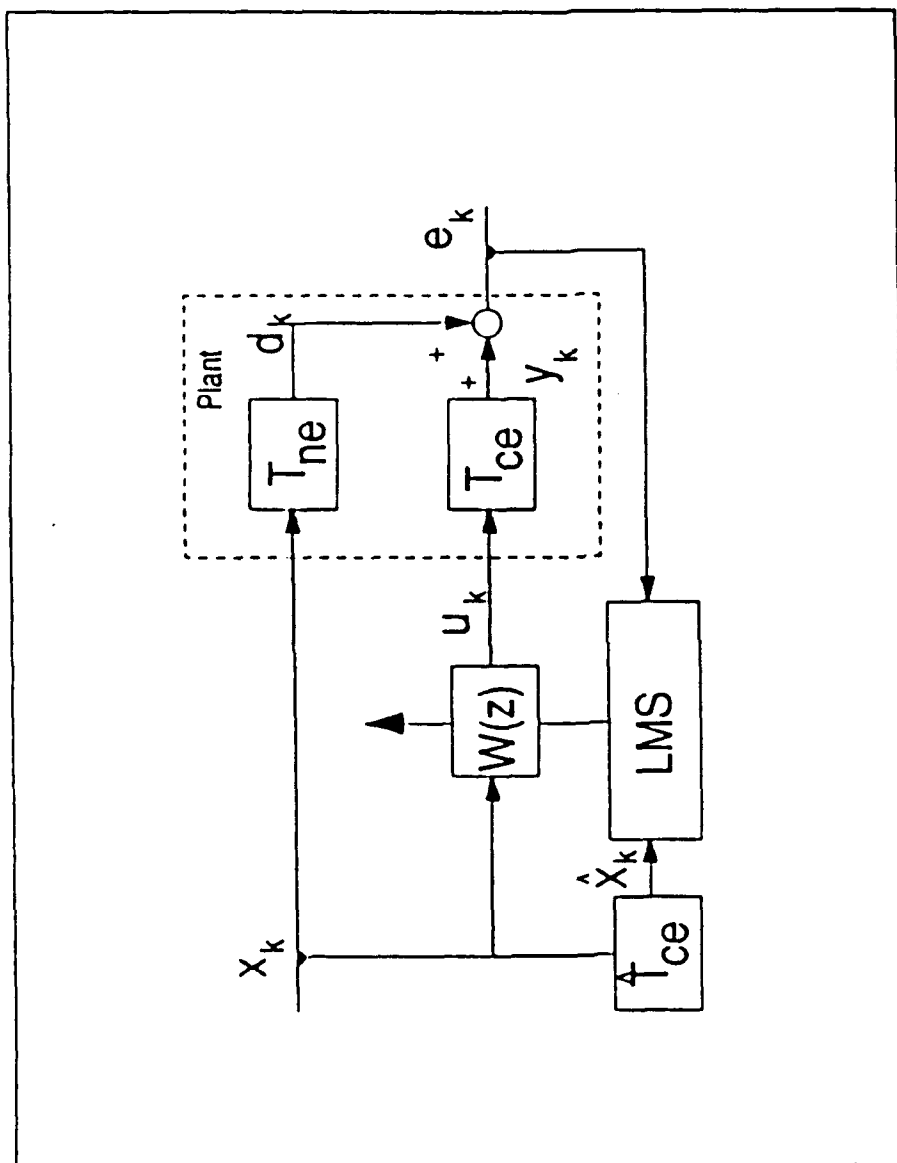


Figure 85. Filtered-x LMS control algorithm for structural vibration control.

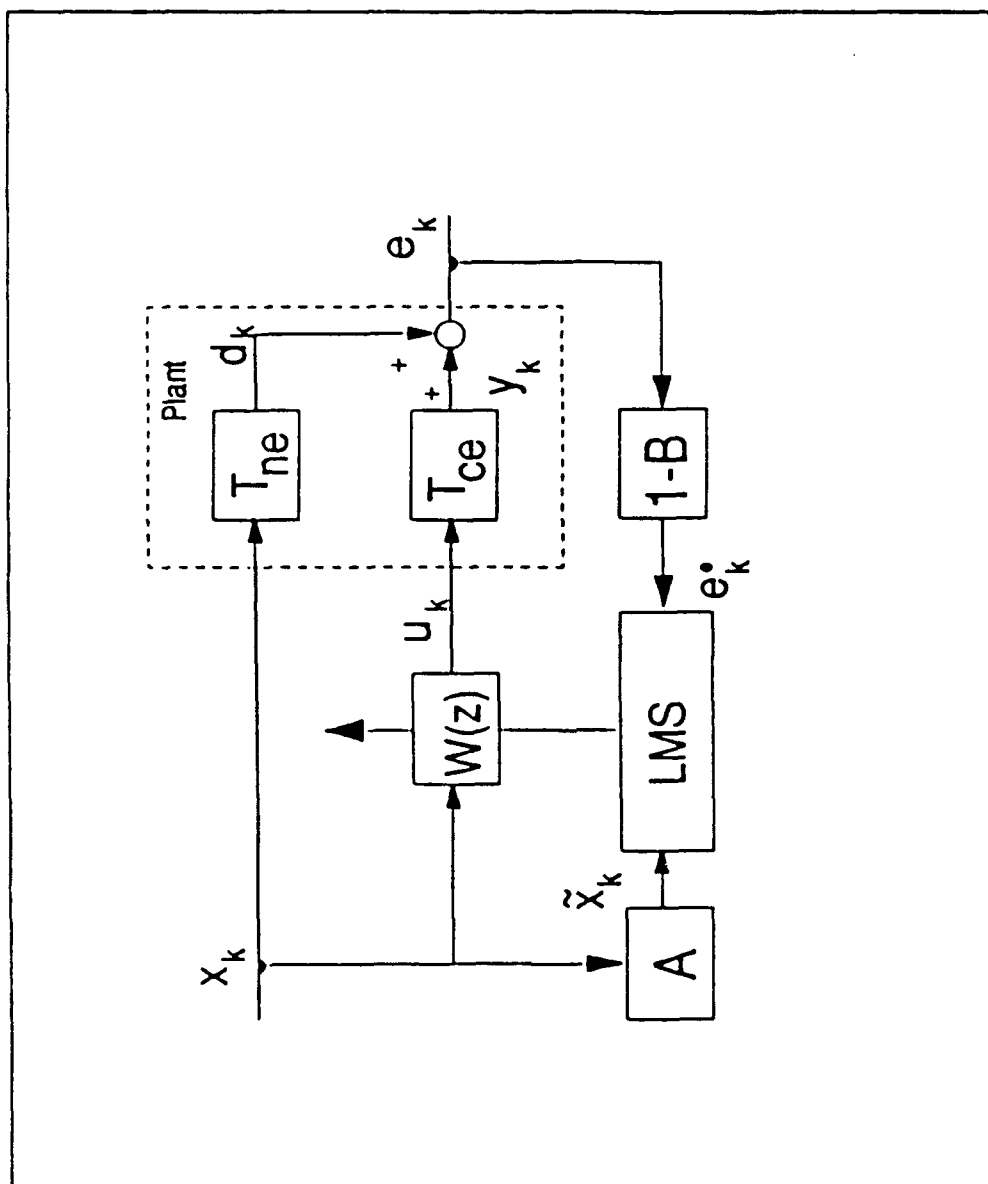


Figure 86. Filtered-x LMS algorithm being used in the equation error control configuration.

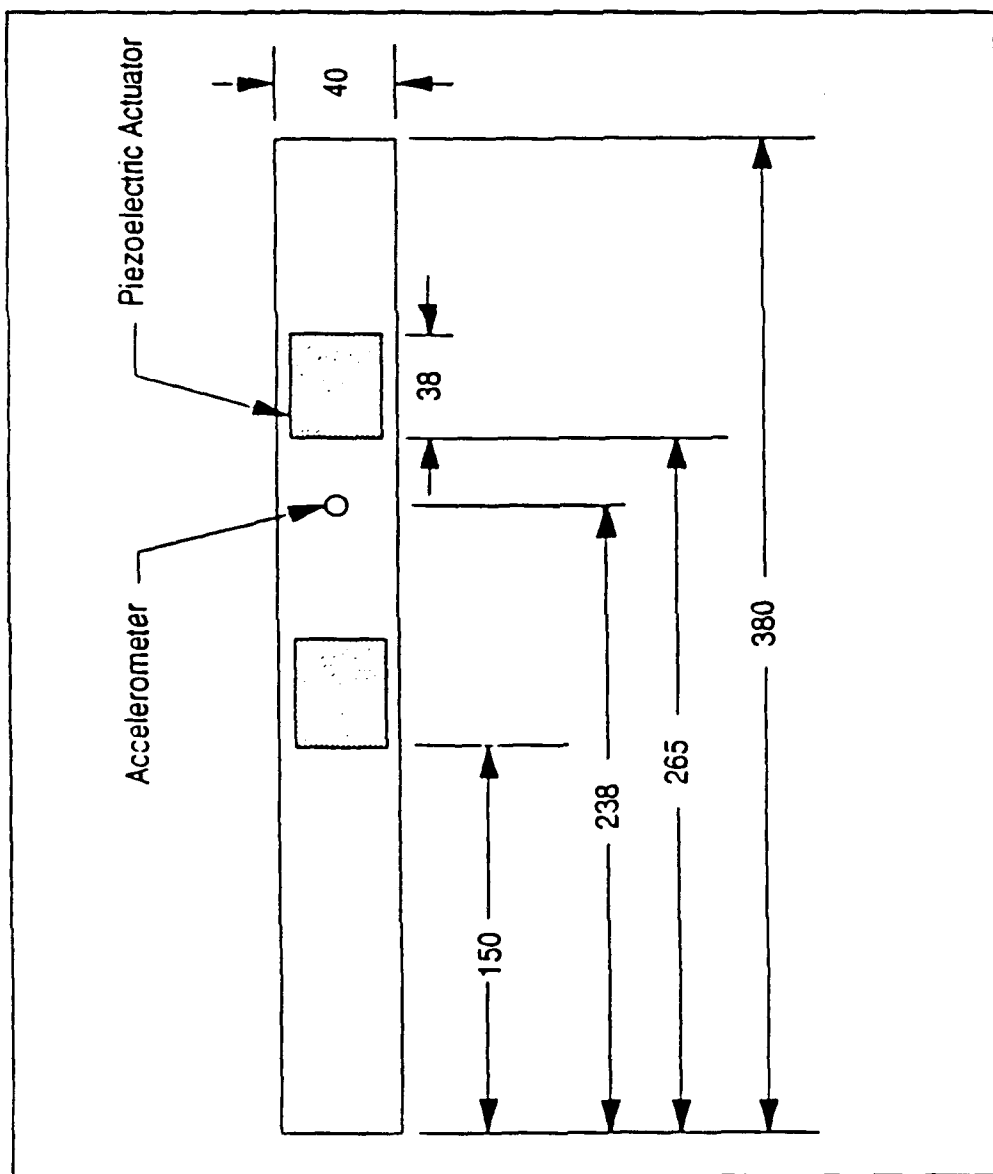


Figure 87. Sensor and actuator locations on the simply-supported beam used for the experimental analysis. All dimensions are in mm.

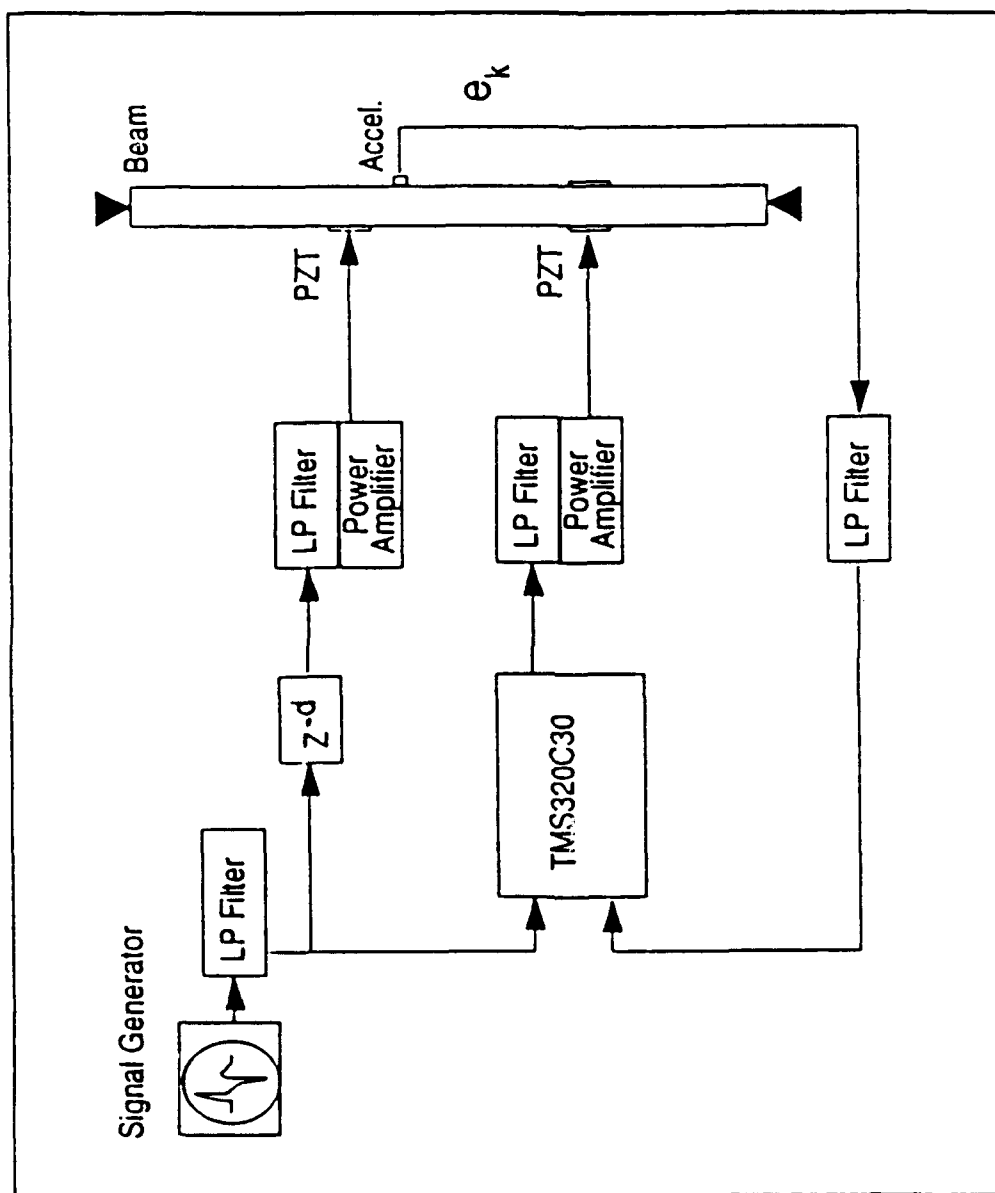


Figure 88. Experimental setup for testing the feedforward broadband control algorithm.

and control experiments was 2000 Hz. As a first step in the experiments random inputs into the control actuator were used for off-line ARMA system identification to solve for the filter coefficients A_i and B_i of $T_{\alpha}(z)$. The ARMA model predicted filtered-x signal can be compared to the real output error signal in the autospectra shown in Figure 89.

Figure 90 shows a comparison of the steady-state error signal measured from the beam both before and after the conventional filtered-x LMS control was applied with a 24th order adaptive FIR and a 5 ms delay in the disturbance. The power spectral density for the two signals in Figure 91 shows the controller attenuating the large frequency components near the structural resonances while adding spillover at the antiresonances. The convergence behavior is disclosed by Figure 92 which shows convergence in about three seconds. The influence of the adaptive filter size on control performance was investigated by varying the filter order from 12 to 80 coefficients. Figure 93 shows the results of these tests for both the casual ($d = 5$ ms), and acasual ($d = 0$ ms). The casual system improves to nearly a constant reduction of 20 dB, and the acasual system is only about 2.5 dB below it.

The Equation Error Control structure was tested with a casual system having 24 adaptive coefficients, and filtering the error signal by the same stabilized poles, $[1-B(z)]$, as used in the previous experiment. Figure 94 shows the steady-state signal results with and without control. The reduction is only 10 dB in power, and the convergence was slower than the conventional filtered-x structure. For a second test, some of the poles of $[1-B(z)]$ were reflected outside the unit circle, and the before and after control steady-state results are shown in Figure 95. The total power reduction is 8.9 dB. Both cases are less than the 20 dB performance from the casual conventional filtered-x control structure. However, this test demonstrates that the Equation Error Control Configuration will work when the IIR filter would be unstable for the Filtered-X LMS Control system.

7.2.3 FEEDFORWARD EIGENPROPERTIES

While there are many papers in the open literature on adaptive feedforward LMS algorithms and approaches to noise control none have addressed the actual mechanisms and characteristics of control inherent in the technique for the control of elastic systems. This research effort has addressed the dynamic characteristics of a feedforward controlled elastic system, and a new mathematical approach to predict the dynamic characteristics of the controlled structure has been developed [73 and 77]. It can be shown that the controlled system effectively behaves as having new eigenproperties. The controlled eigenvalues and eigenfunctions are a function of the control force and error sensor locations, and independent of the input disturbance. Numerical analysis demonstrate the applicability of the proposed formulation, and these results have been corroborated experimentally.

The applicability of the formulation is demonstrated [73] for a uniform simply supported beam as shown in Figure 96. The beam is made of steel and has a bending stiffness $EI = 1857 \text{ Lb.in}^2$, mass per unit length $m = 0.000089 \text{ lb.sec}^2/\text{in}^2$ and length $L = 14.96 \text{ in}$. For analysis the beam is assumed to have 0.1% damping in each mode, and ten modes are included in the response

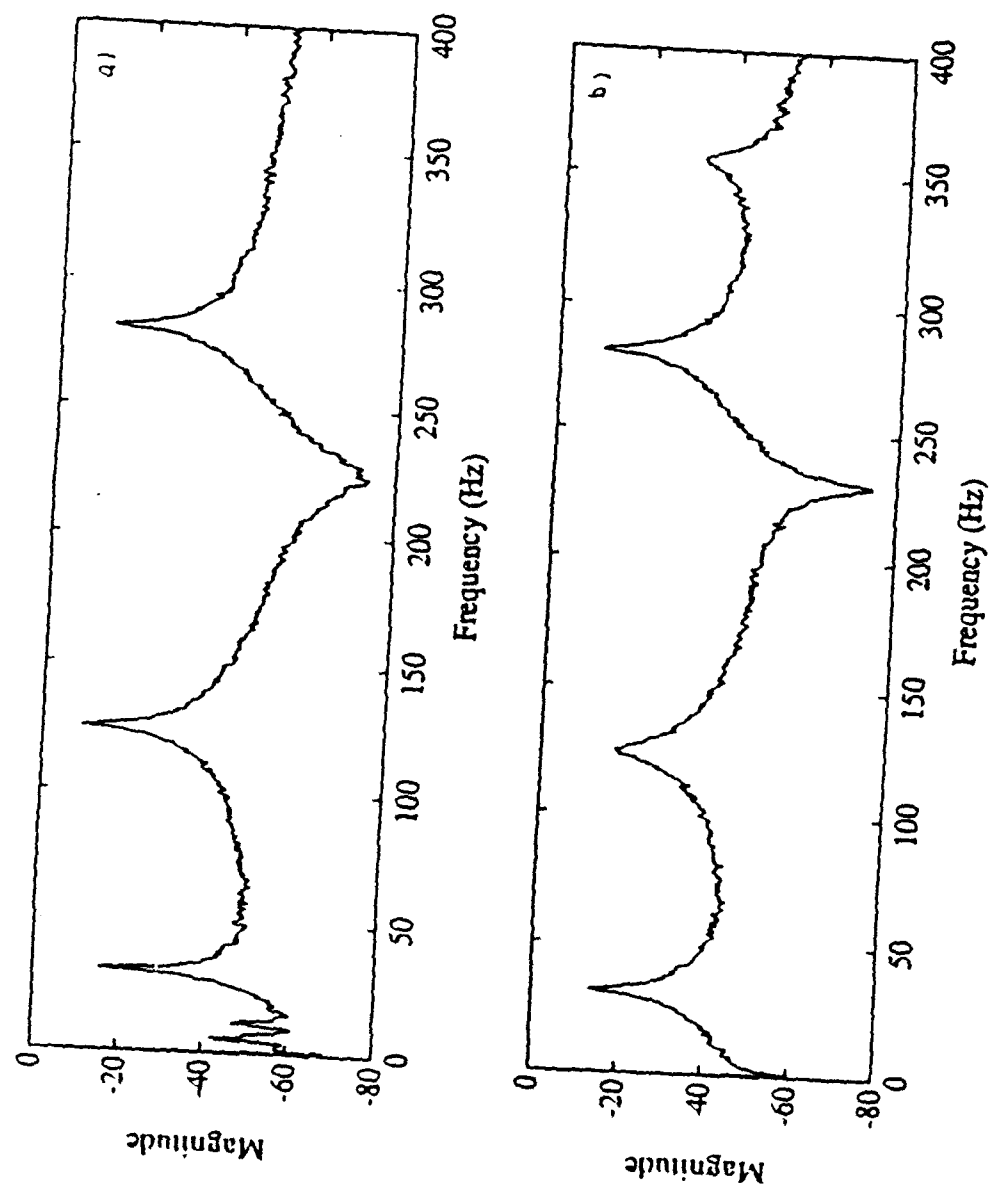


Figure 89. Autospectrum of the control-loop response when excited by white noise.
(a) measured; (b) predicted filtered-x signal.

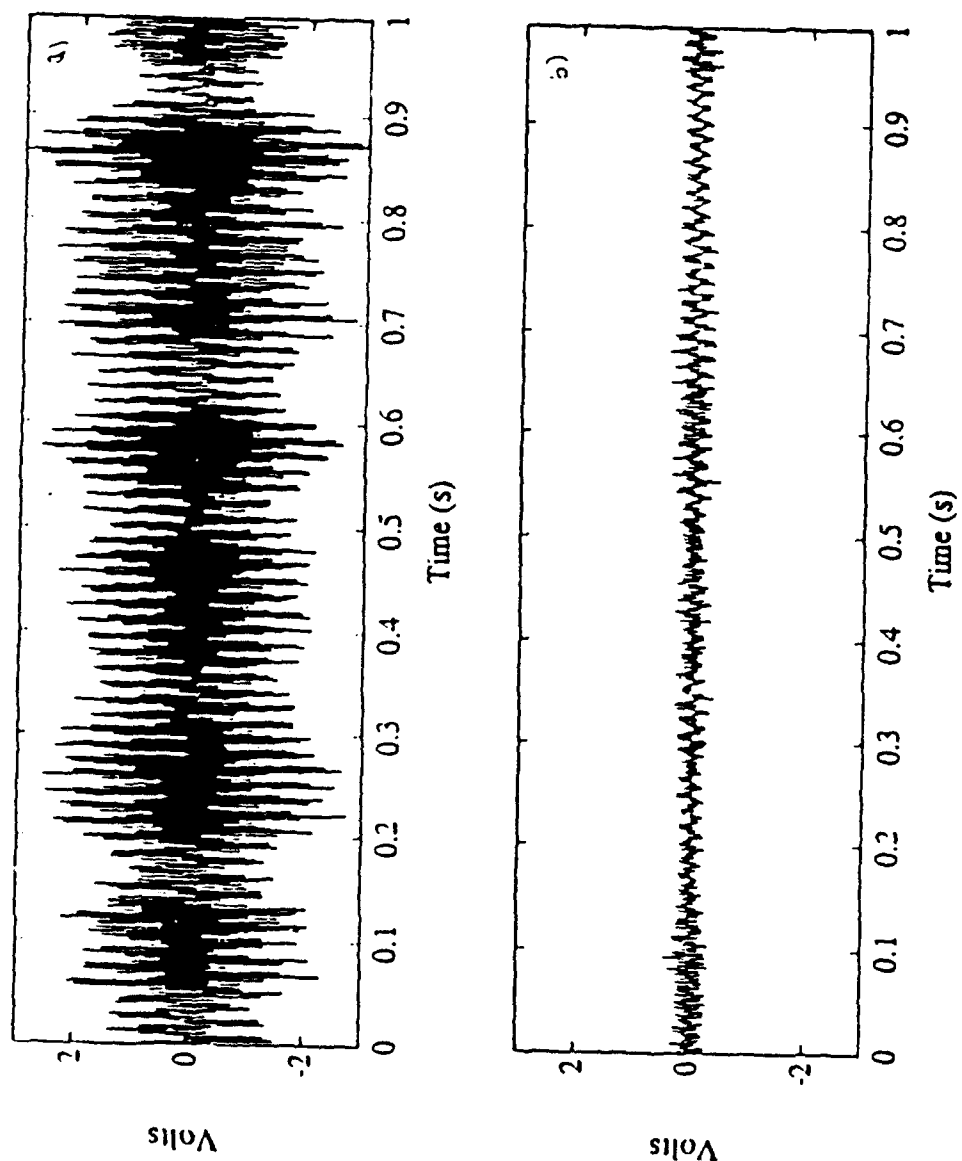


Figure 90. Error signal from the plate. (a) before control; (b) after control.

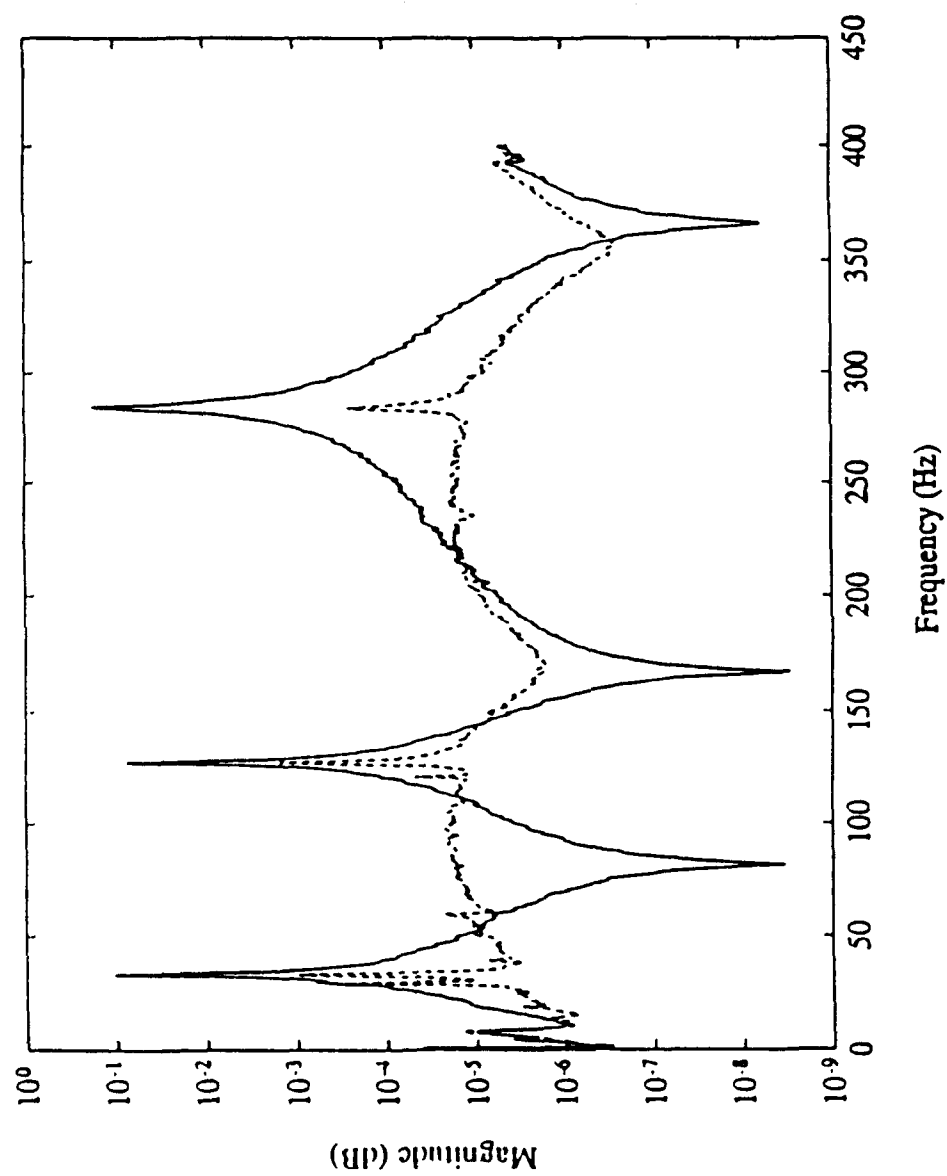


Figure 91. Autospectrum of the error signal.
 — before control; — after control.

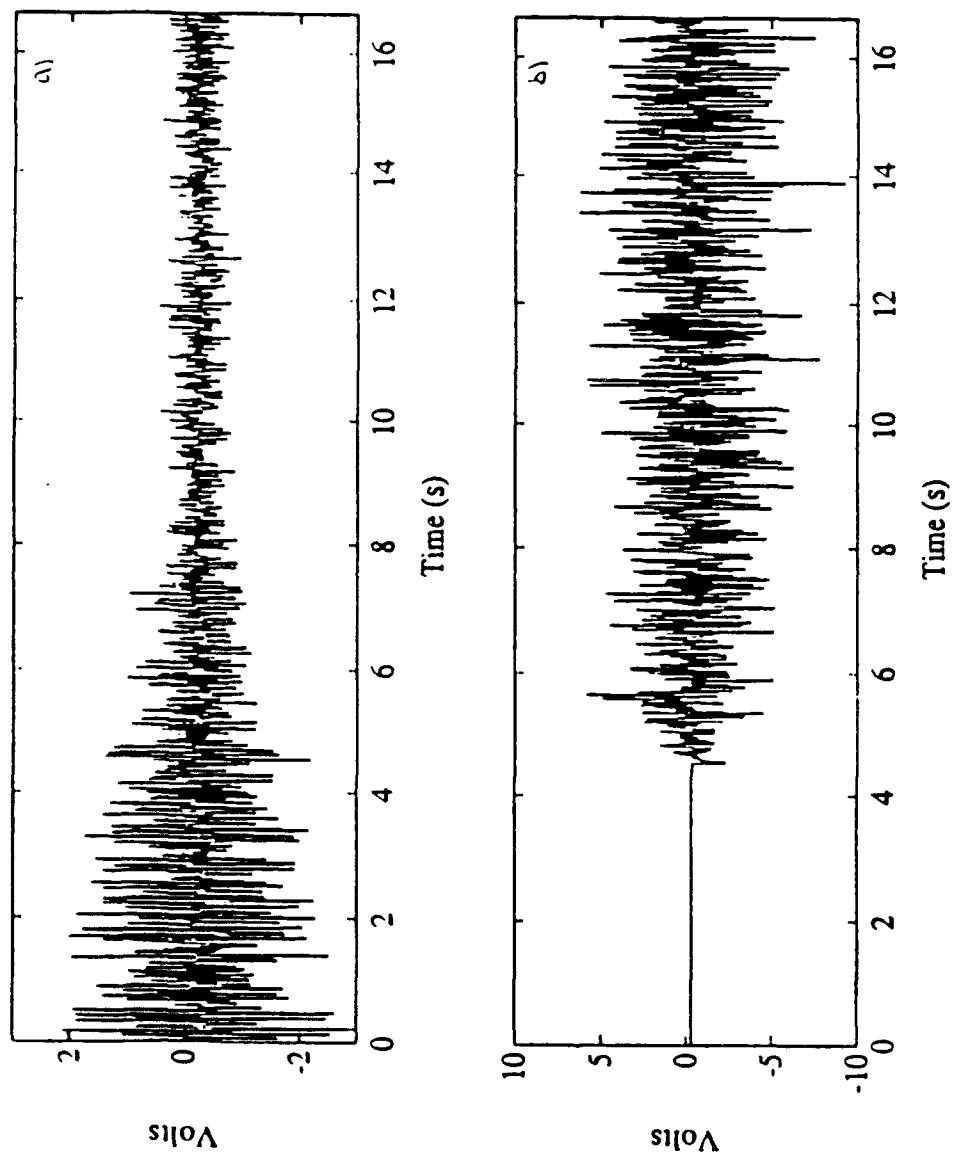


Figure 92. Time histories showing the convergence rate of the adaptive controller.
 (a) error signal; (b) control input signal.

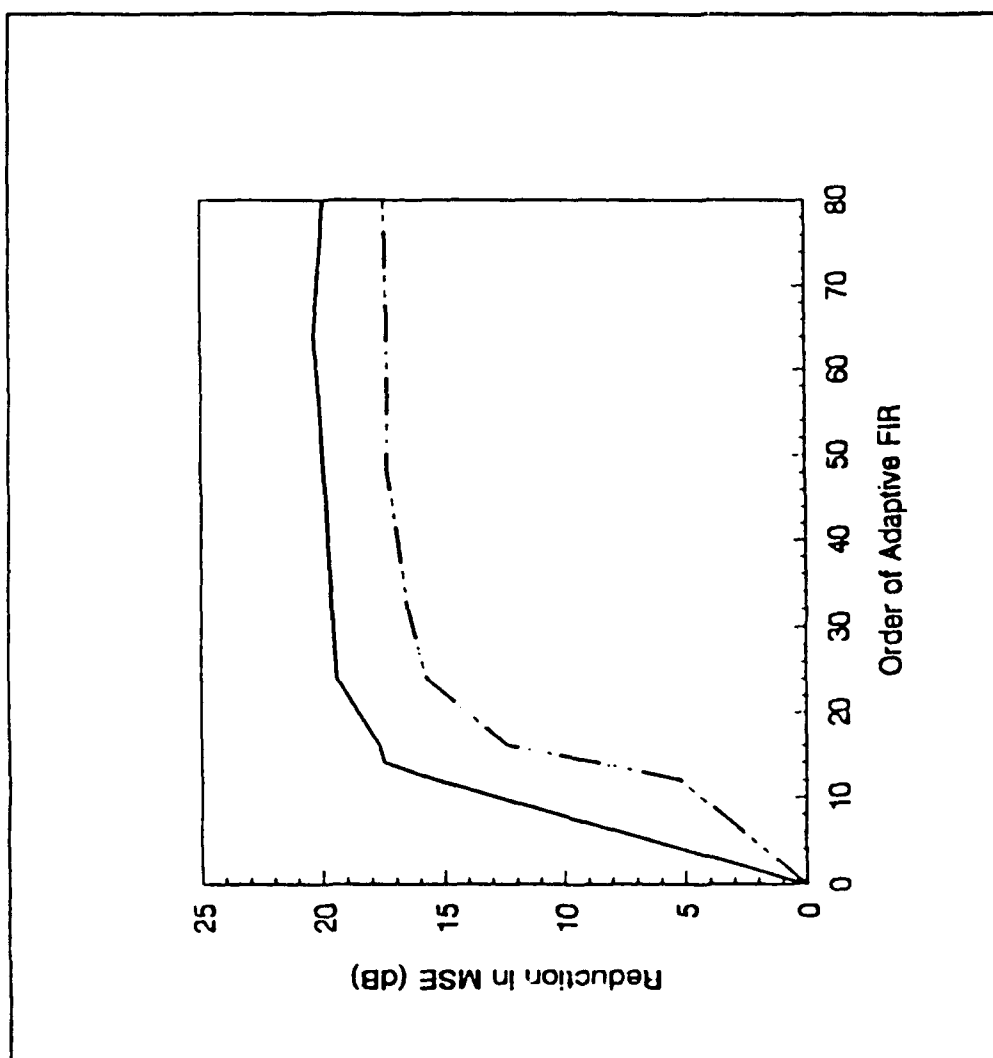


Figure 93. Performance of the filtered-x LMS controller as the filter size is increased.
 — casual; - - - acausal.

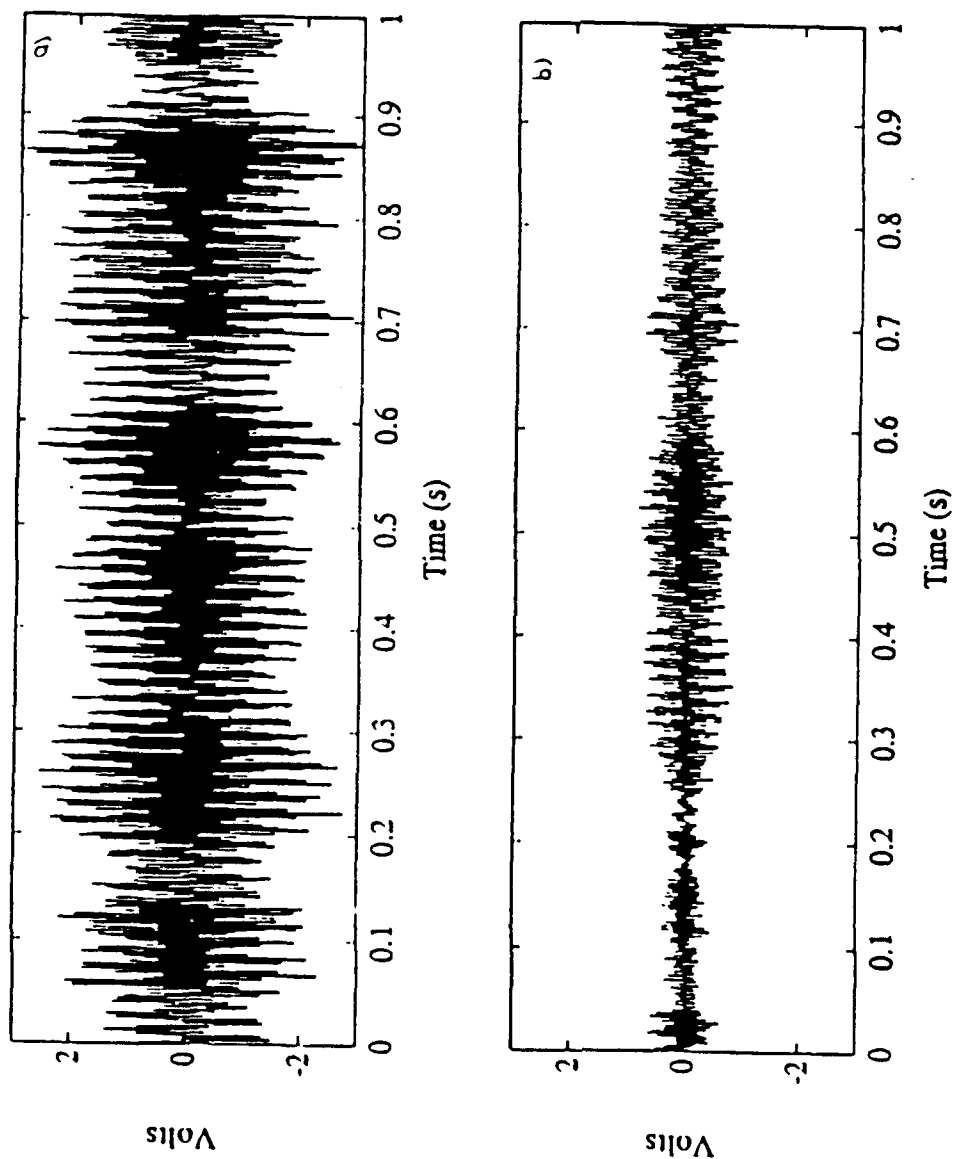


Figure 94. Actuator error signal time histories during application of the equation error configuration.

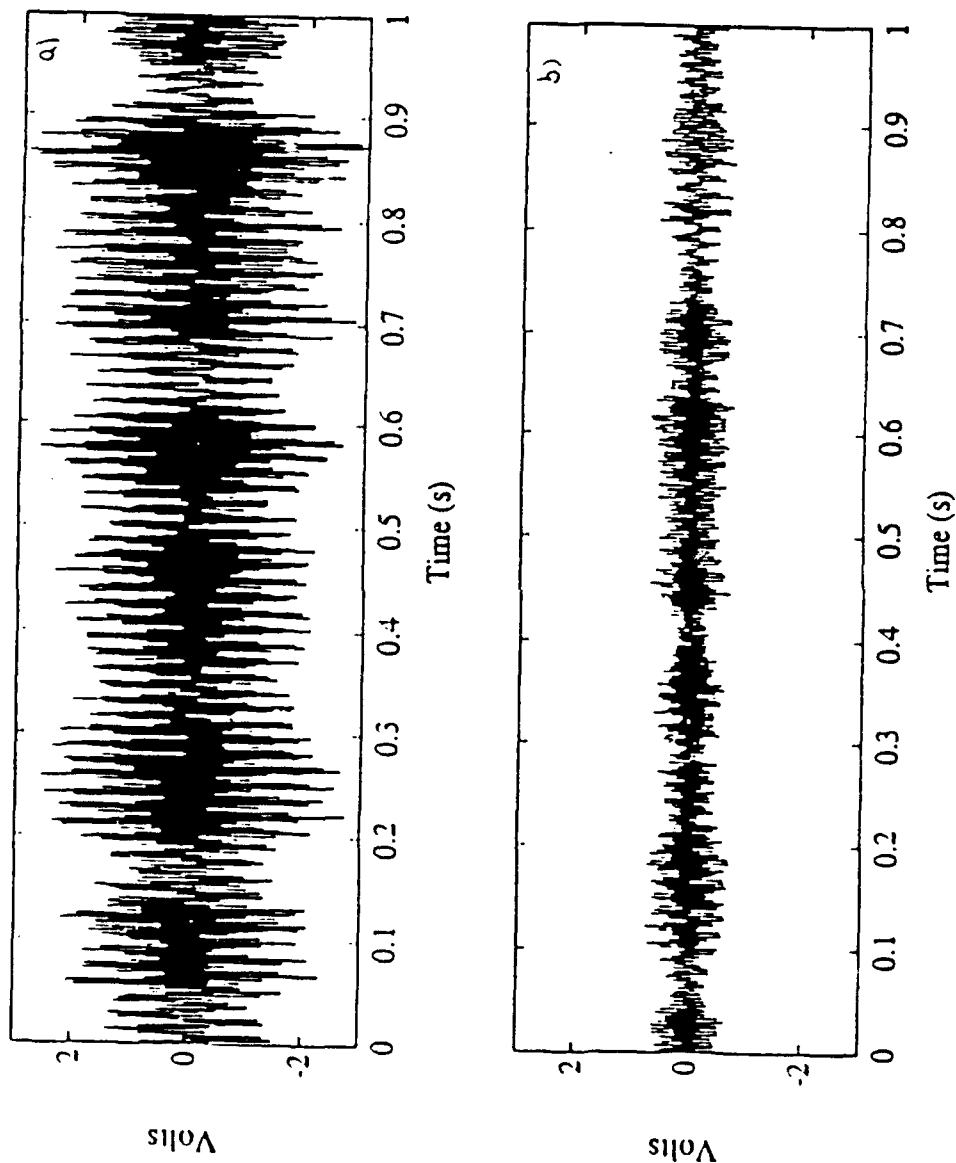


Figure 95. Error signal time histories showing effects of applying the equation error control form with unstable "poles". (a) without control (b) equation error control configuration.

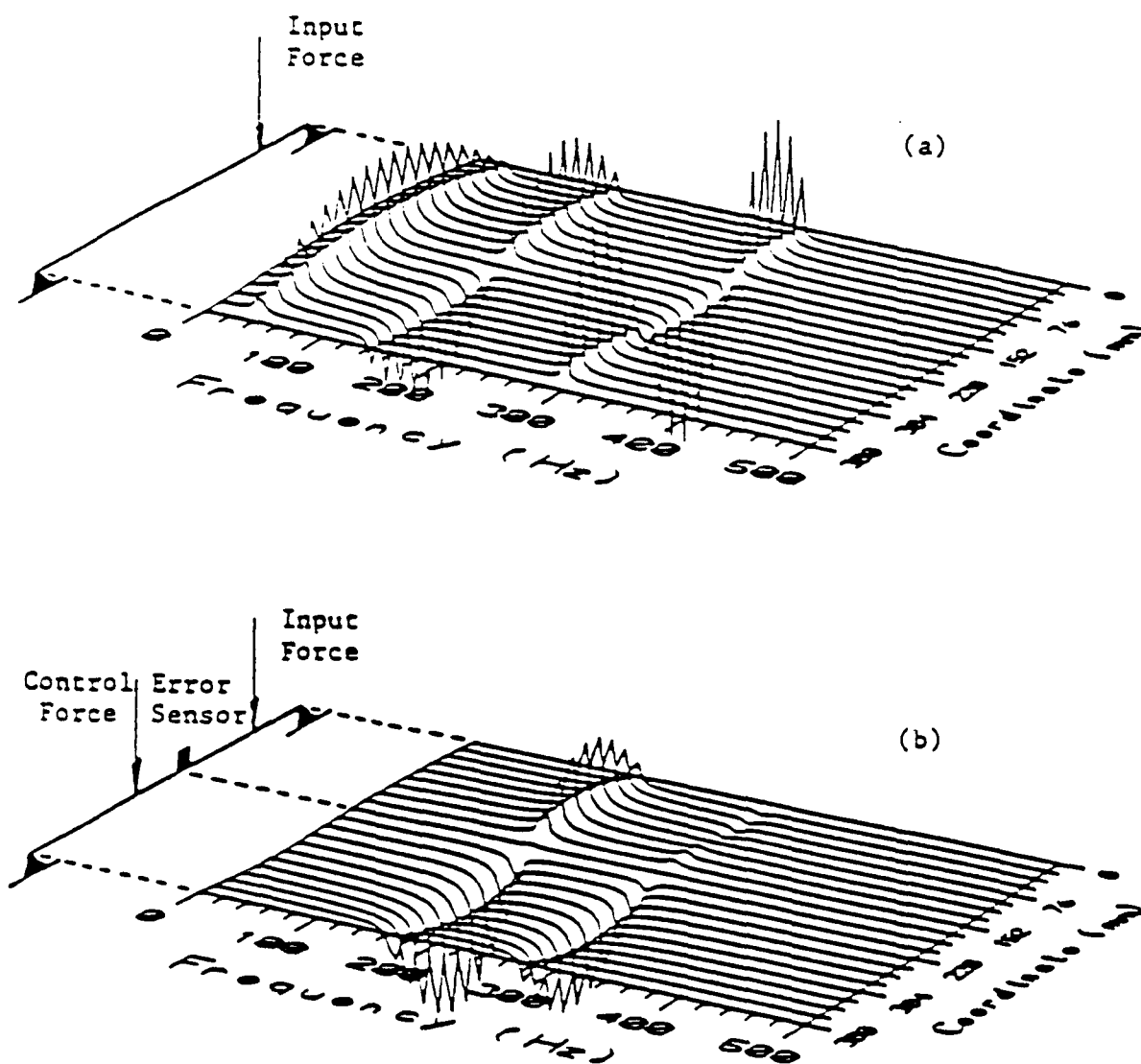


Figure 96. Beam frequency response curve (a) uncontrolled; (b) controlled.

analysis. The beam and controller experiment setup are shown in Figure 97, and the filtered-x LMS adaptive control algorithm with two coefficients was implemented in the TMS320C20 DSP. The beam is excited by a concentrated force located at $a = 0.1L$, and the control force is placed at $b = 0.65L$. The error sensor is located at $c = 0.45L$. Assuming a unit magnitude of input force, Figure 98 shows the amplitude and phase of the displacement at $x=a$ as a function of frequency. It is clear that the controlled system behaves as having new eigenproperties different from the original uncontrolled beam modes corresponding to the resonances. The controlled system eigenvalues are obtained by solving for the roots of a polynomial, which is the characteristic equation of the controlled structure. The mode shapes associated with the controlled natural frequencies were computed as a linear combination of the original or uncontrolled beam eigenfunctions. The first three eigenfunctions are shown in Figure 99(a) through 99(c), while Figure 100(a) through 100(c) present the third, fourth and fifth mode shapes. As can be seen, all eigenfunctions have a nodal point at the error sensor location. Thus, control is achieved by creating new modes which all have zero response at the error sensor location. This view should be contrasted with the conventional interpretation based on the uncontrolled modes "cancelling" at the error point so that their superposition is zero.

This eigenanalysis work has been extended to a design approach for shaped PVDF distributed sensors to work with a feedforward controller to modify the dynamic properties of a structure-controller system [77]. Thus a distributed error sensor can be designed to induce the desired dynamic characteristics of the system. In Ref 77 a feedforward controller is designed to modify the dynamic properties of a simply supported beam so the beam-controller system has the modal characteristics of a beam-spring-mass system. The resulting sensor shape is illustrated in Figure 101, and the changes in the system resonances are clearly seen in the acceleration response plot in Figure 102. Here the uncontrolled response is simply that of a beam, and the controlled response is that of a beam-spring-mass system. Thus the controller has been designed to make the uncontrolled system behave in a required manner as a totally new system.

7.2.4 MODEL REFERENCE CONTROL

A model reference active structural acoustic control approach was investigated as a method for replacing microphone error sensors located in the acoustic field with accelerometers located on the structure [78]. The controller is the multi-input/multi-output adaptive LMS algorithm as illustrated in Figure 103. As opposed to driving the response of the sensor locations on the structure to zero with the control inputs, the response is driven to some pre-determined "reference" value corresponding to a cost function which originally implemented microphones or the supersonic region of the wavenumber transform as error information. In effect, the uncontrolled structure is adaptively modified to behave like the reference structure. Results indicate that the identical optimal control solution can be obtained when implementing the structural sensors, and the required number of sensors must simply equal the number of control actuators. Figure 104 shows a simply supported beam that is used for the analytical analysis, and it is assumed to be located in an infinite baffle. The one dimensional beam model includes the material properties for steel with a point disturbance force driven at 400 Hz (off-resonance), and three piezoelectric actuators. In the first test case, the supersonic region on the κ_x axis of

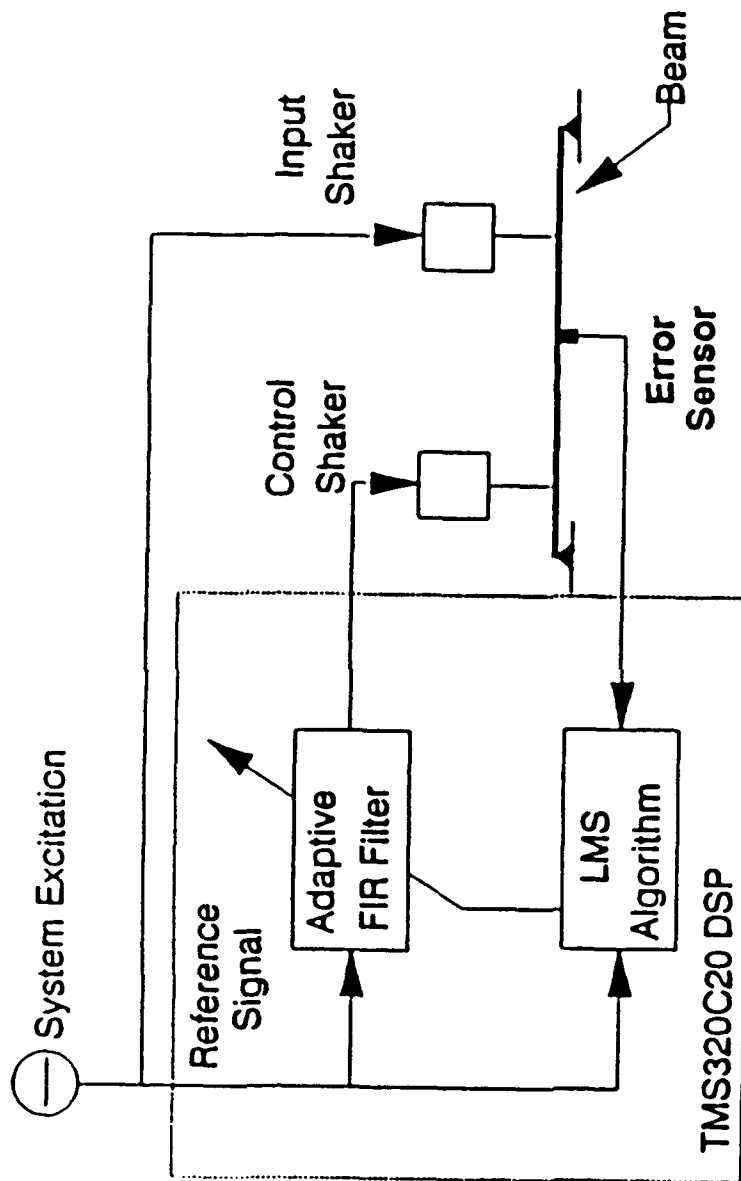


Figure 97. Time domain LMS control block diagram.

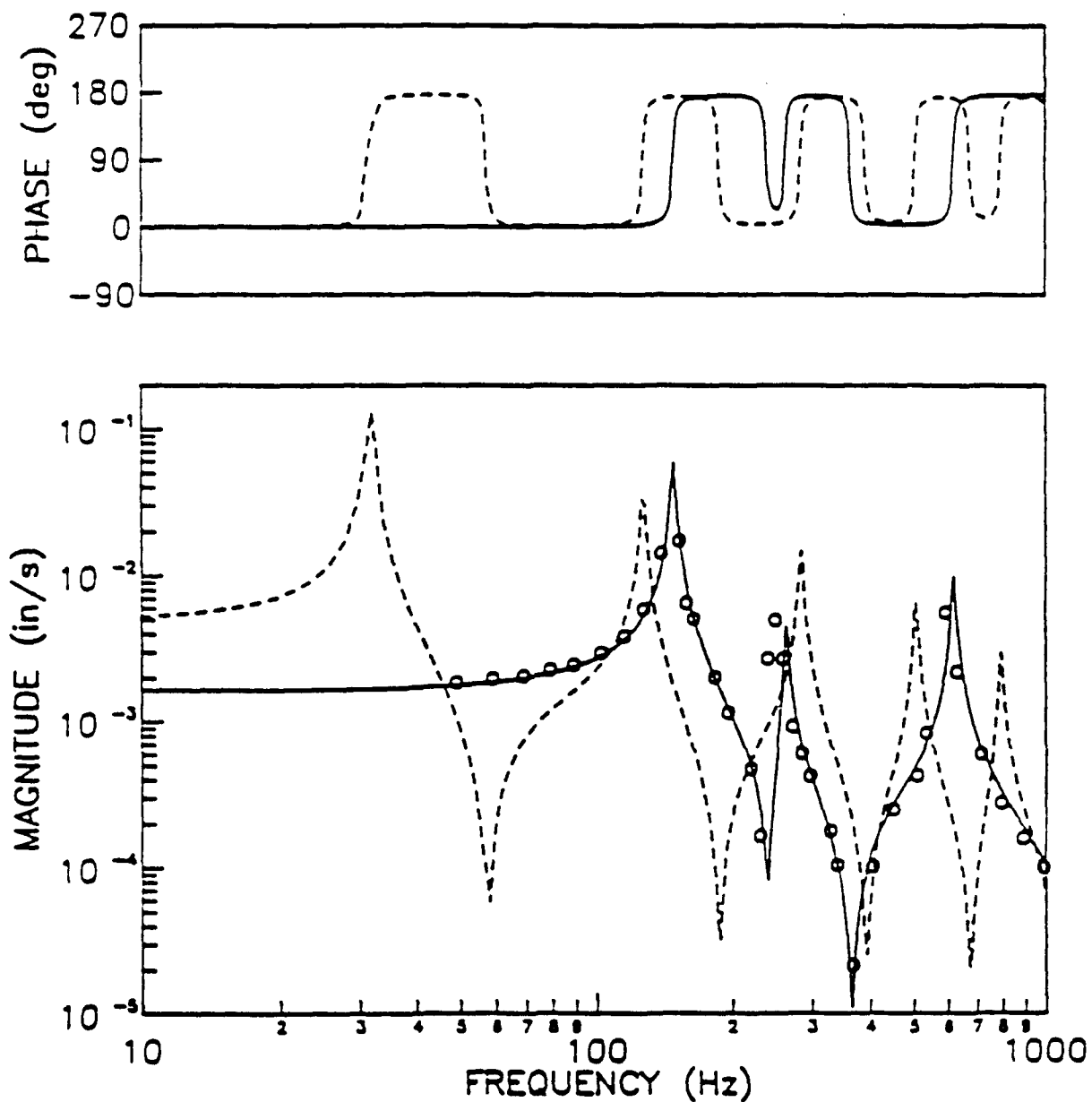


Figure 98. Displacement response at input force location vs. excitation frequency.
 —, uncontrolled analytical; —, controlled analytical;
 ○ ○ ○, controlled experimental.

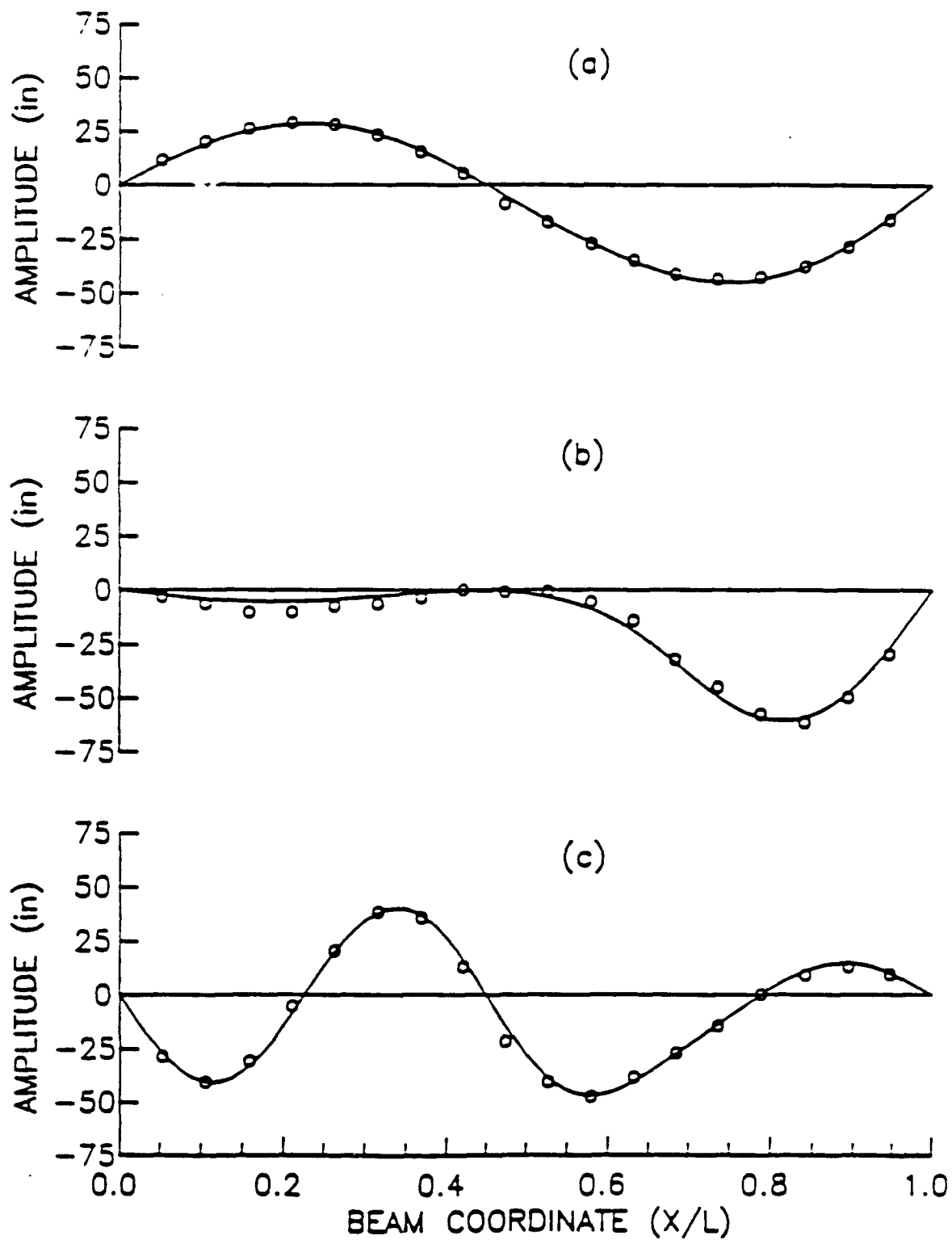


Figure 99. (a) First, (b) second and (c) third controlled system mode shapes. —, analytical; $\circ\circ\circ$, experimental.

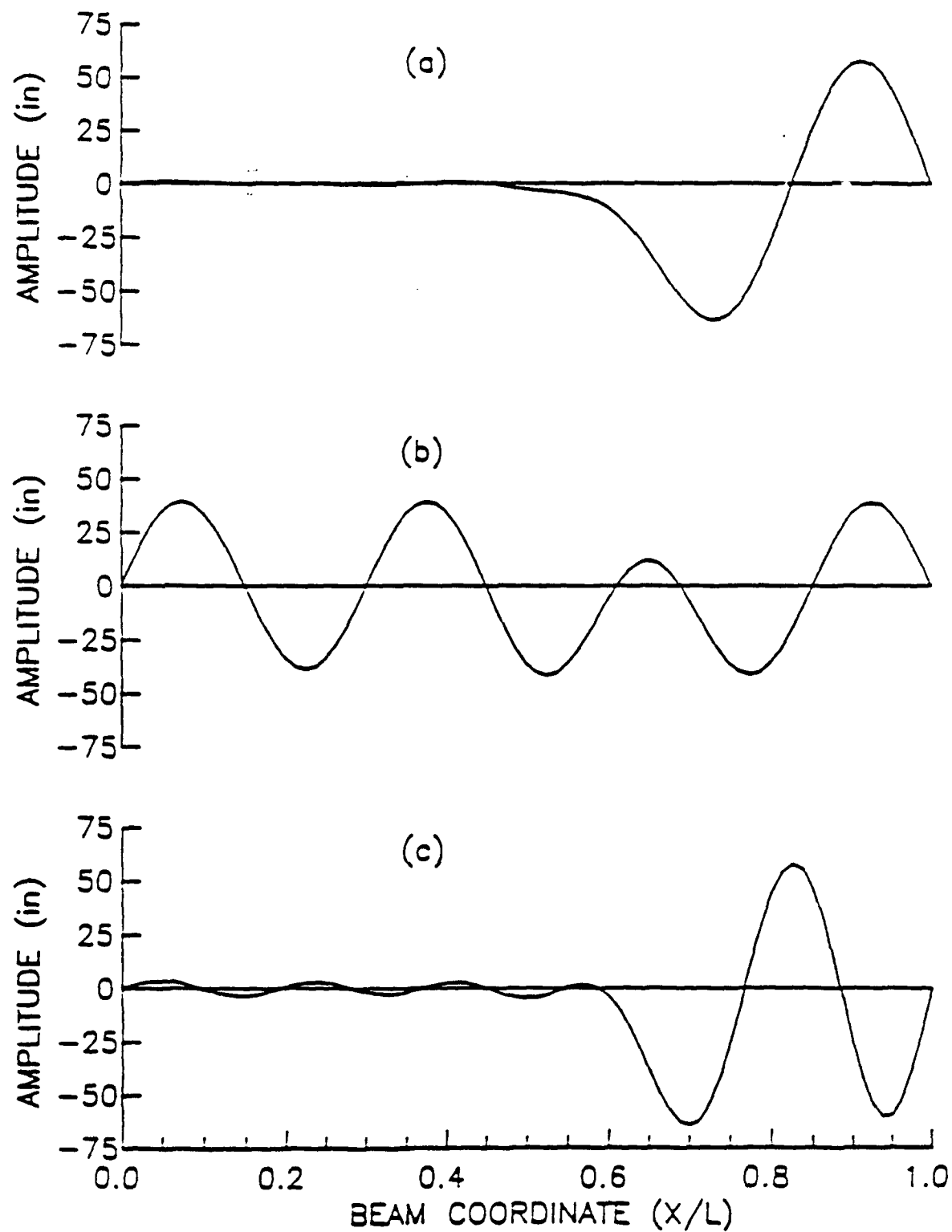


Figure 100. Analytical (a) fourth, (b) fifth and (c) sixth controlled system mode shapes.

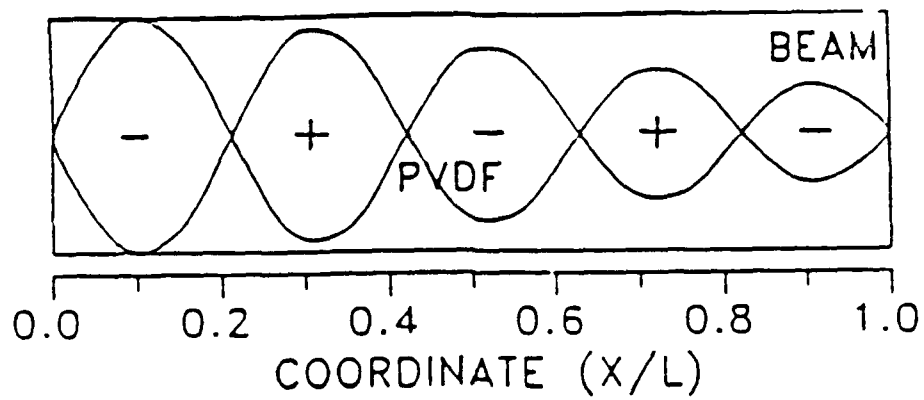


Figure 101. PVDF error sensor for beam-mass example problem.

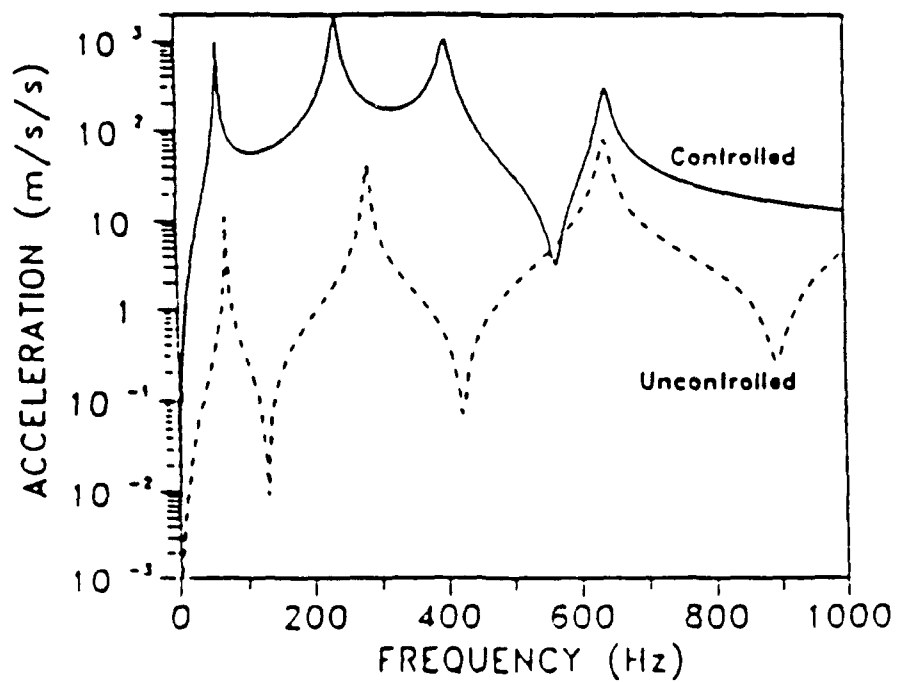
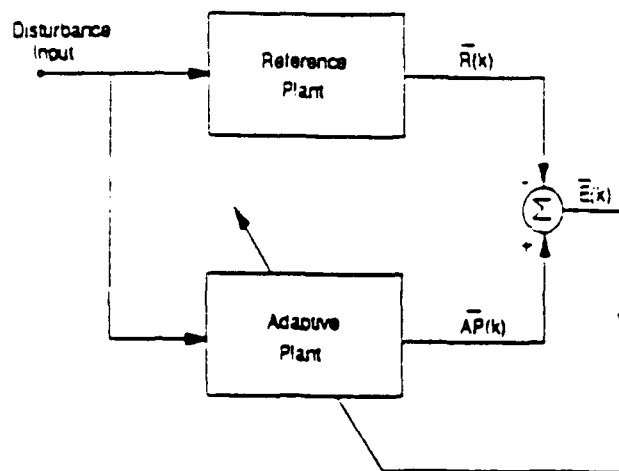
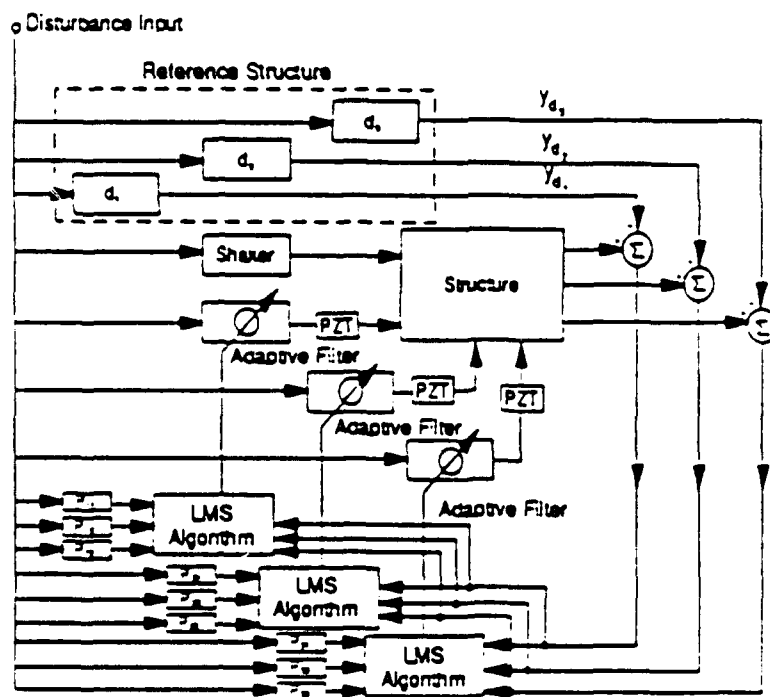


Figure 102. Acceleration response at disturbance force location for example problem.



(a) Conceptual Schematic of Model Reference Controller



(b) Detailed Schematic of Model Reference Controller

Figure 103. LMS model reference controller schematic.

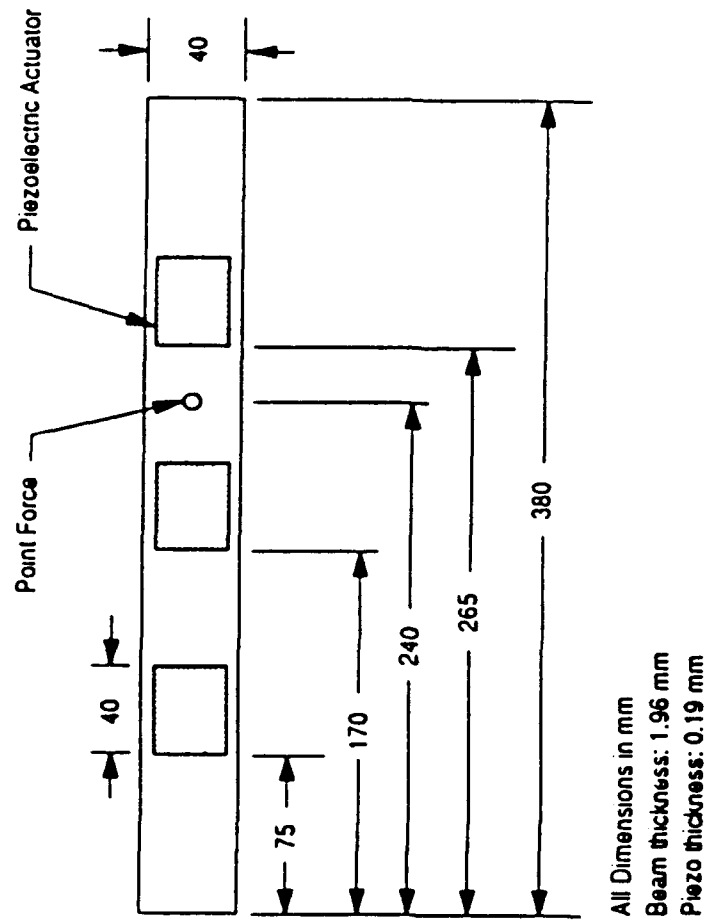


Figure 104. Schematic of simply supported beam.

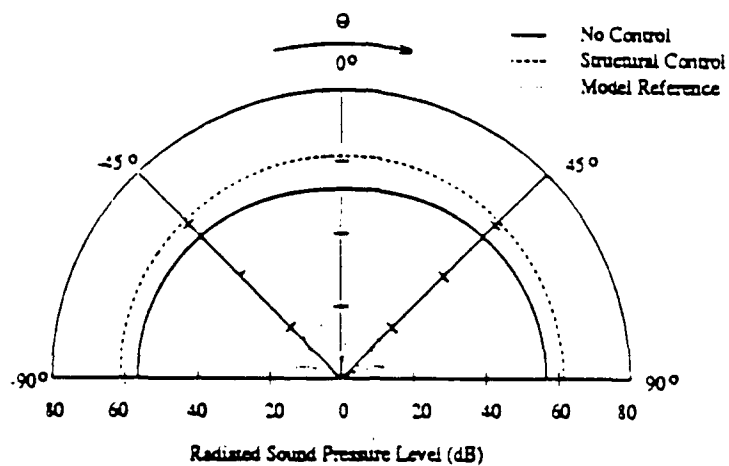
the wavenumber domain was chosen as the initial cost function, and the complex structural response at three axial locations of 84.4 mm, 126.7, and 168.9 mm was computed. These responses are used to implement the reference model. The far-field radiation directivity and spectral amplitudes of the wavenumber transform of the structure are illustrated in Figure 105. For comparison purposes, the results for the structural response being driven to zero at the chosen sensor positions is shown as well, and the radiation actually increases. In the reference model approach the, the wavenumber spectrum is minimized significantly in the supersonic region while in the structural control approach this region is increased. For the second case, three widely separated microphone positions at a constant radius of 1.2m were implemented as error sensors in the cost function, and the residual structural responses at the previous used locations was used as the reference model. As indicated in Figure 106, the far-field acoustic radiation from the beam was attenuated by approximately 40 dB, and the supersonic region was reduced significantly. This test case was chosen to emphasize that while the k-transform for complex structures may not be readily obtainable, the acoustic can be measured to design a reference model. In addition, for the example studied here, significant reduction in far-field sound radiation can be achieved even when errors of $\pm 5\%$ are present in the reference model as shown in Figure 107. A considerable level of attenuation in sound can be achieved for a band of frequencies within $\pm 5\%$ of the reference frequency used to obtain the models for creating the disturbance as shown in Figure 108. Thus it appears that the model reference approach is reasonably robust.

An interesting aspect of the LMS approach is that there is much similarity between its arrangement and the original Rosenblatt "perceptron" system which is the fundamental basis of artificial neural networks [79]. Thus adaptive LMS filters can also be viewed from the artificial neural network approach as devices which are trained by various methods to model the system plant. When the noise is minimized (for a single input/single output system) the FIR coefficients will contain information related to the plant transfer function. This observation implies that much of the progress presently being achieved in neural networks may be soon implemented to feedforward LMS techniques to create "smart" controllers, particularly for control of non-linear systems as discussed next.

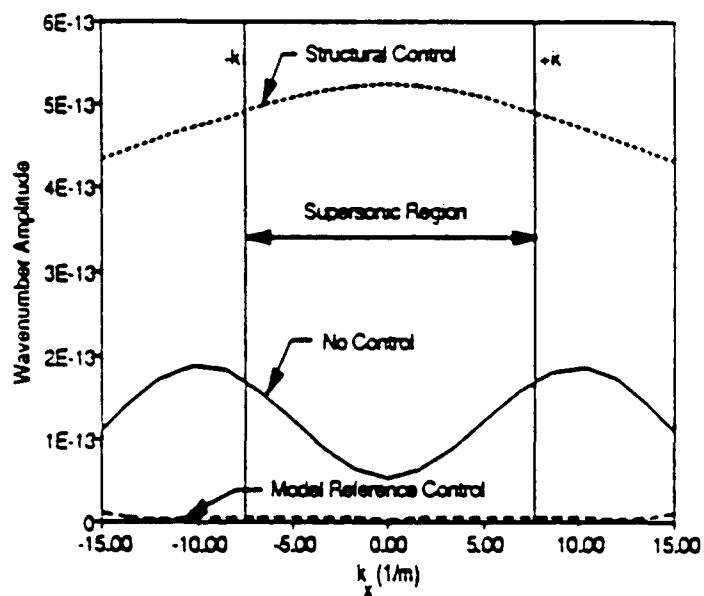
7.2.5 NEURAL NETWORKS

Adaptive LMS feedforward control has demonstrated much success in attenuating sound and vibration in a variety of difficult applications. However, all applications of LMS control have been limited to linear systems, and the problems of nonlinear response of dynamical systems remain largely unresolved. Artificial Neural Networks (ANN) have demonstrated much potential for system identification and classification in a variety of nonlinear problems. The potential for using ANN in active control of sound and vibration as an extension of linear feedforward control approaches (adaptive LMS) has been examined through simulations and experiments at VPI&SU [80].

The block diagram for an ANN is shown in Figure 109 the disturbance is pass through the nonlinear plant producing a plant output, and used as a reference for the ANN to produce the

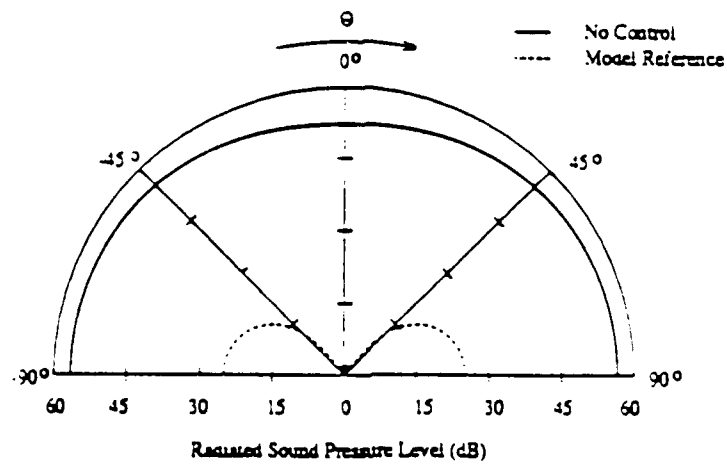


(a) Radiation Directivity

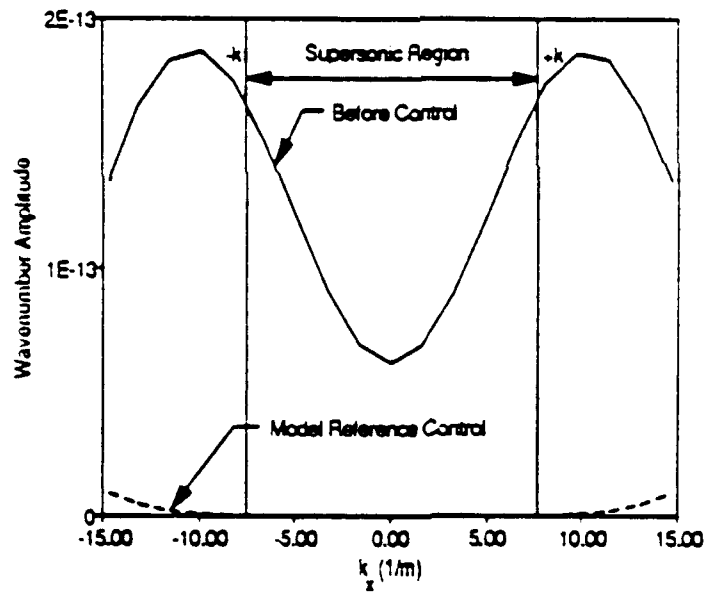


(b) Wavenumber Distribution

Figure 105. Model Reference control implemented from wavenumber transform error data, $f = 400$ Hz.

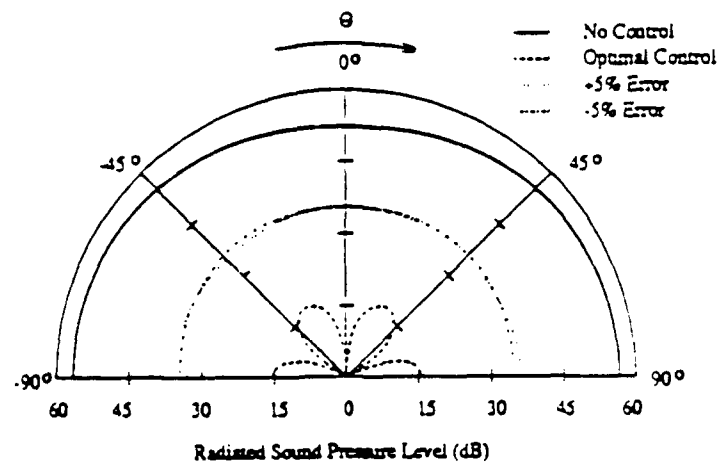


(a) Radiation Directivity

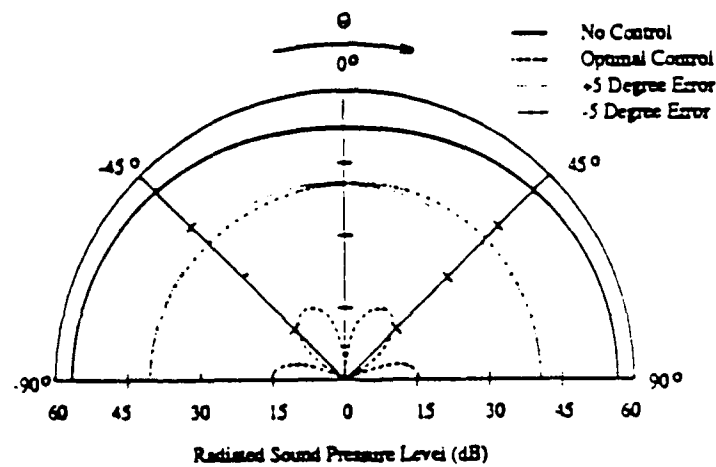


(b) Wavenumber Distribution

Figure 106. Model Reference control implemented from microphone error data, $f = 400$ Hz.

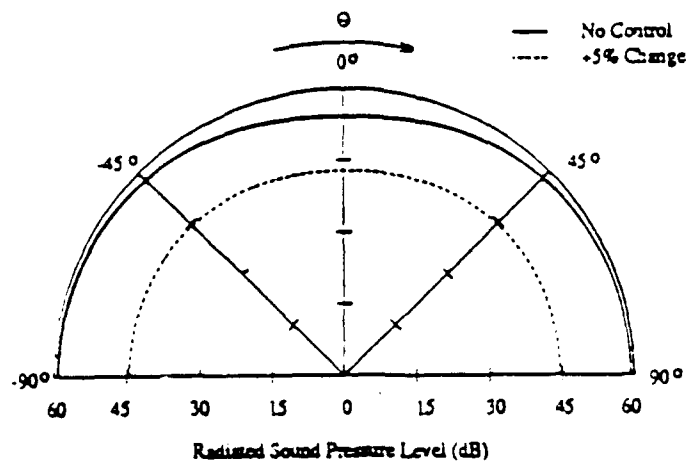


(a) Error in Magnitude

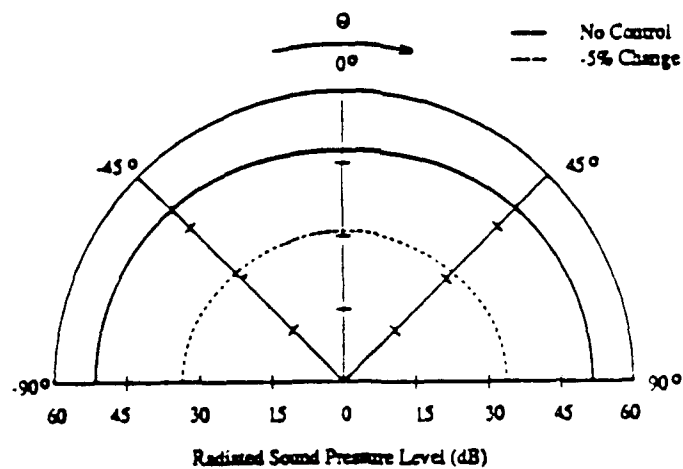


(b) Error in Phase

Figure 107. Model Reference control implemented from microphone data
 $f = 400 \text{ Hz}$.



(a) Reference Frequency of 380 Hz



(b) Reference Frequency of 420 Hz

Figure 108. Model Reference control implemented from microphone data with shift in reference frequency.

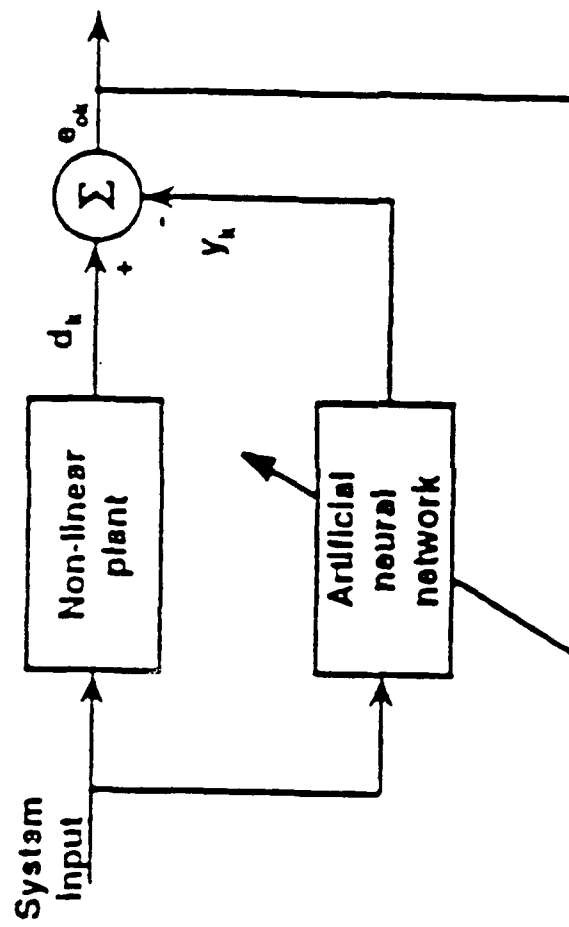


Figure 109. Artificial neural network (ANN) application functional block diagram.

control output. The objective is to minimize the difference between the plant and control outputs. Figure 110 shows the layout of the multi-layered ANN where the reference (or training signal) is sampled by six tap delay points and feedforward through adaptive weights to the three hidden layer neurodes. The inputs to each neurode are summed and passed through a nonlinear squashing function to produce the neurode output. These outputs are passed through another set of weights, and then summed to produce the ANN output. The output of ANN can be a nonlinear function of the inputs due to the squashing functions.

To demonstrate the ANN approach the plant used in the experiment was an overdriven amplifier producing a clipped out when driven by single frequency sinusoid. The ANN controller was implemented on a TMS320C25 DSP resident in a PC. Figures 111 and 112 present the results of the simulation and experiment, respectively. Attenuations on the order of 24 dB of rms value for the experiment and 27 dB for the simulation are apparent. These results show that the ANN has much potential in the implementation of feedforward controllers for nonlinear systems.

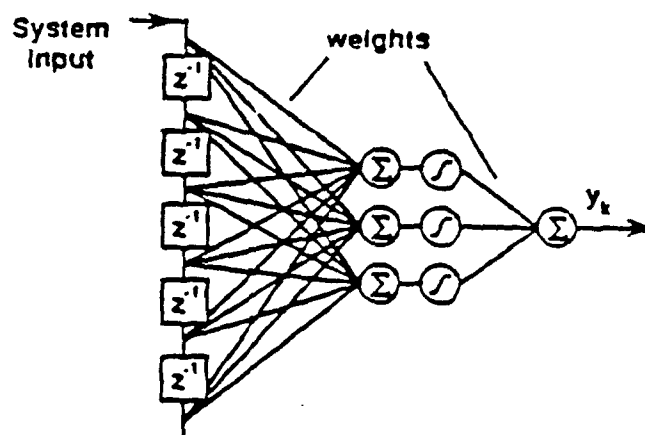


Figure 110. ANN configuration.

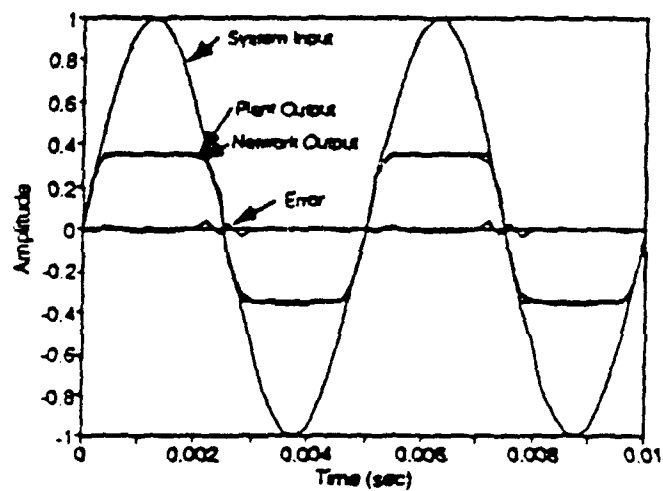


Figure 111. ANN simulation results.

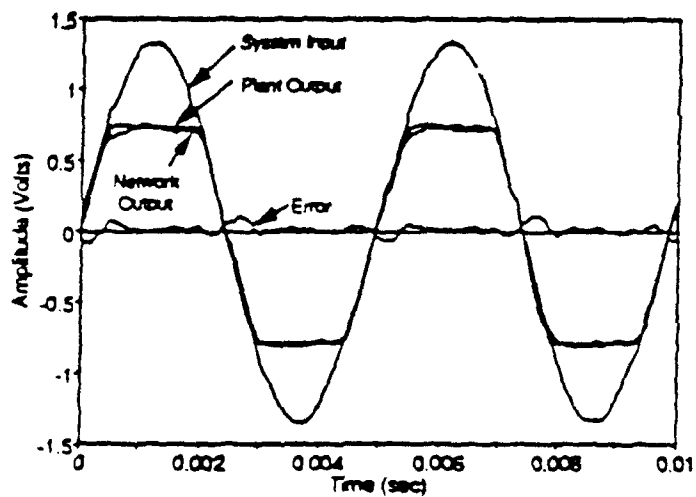


Figure 112. ANN experimental results.

SECTION 8.0

TECHNOLOGY TRANSFER

This section provides a short discussion of the issues in transferring the developed technologies to real world and practical applications.

With partial support from a separate ONR contract a conference was held at VPI&SU on recent advances in active acoustics in April 1991. The conference was attended by over 350 researchers and interested professionals from academia, industry, and government. A total of 70 papers were presented by U.S. and foreign researchers, and this Grant supported the presentation of 5 papers by VPI&SU. This conference clearly demonstrates the level of activity in this rapidly advancing technology. It additionally provided VPI&SU the opportunity to exhibit and demonstrate ongoing DARPA/ONR research, and conduct working level technical exchanges with a large number of other researchers and potential users of the technology. The industrial interest in the active structural acoustic research at VPI&SU was exceptional.

VPI&SU has worked during this Grant period with a number of defense contractors with serious interests in the practical application of active structural acoustic control and adaptive materials to real world problems. This has included funding from Westinghouse, Dupont, Mechanical Technology Incorporated and Noise Cancellation Technology for specific technology transfers and developments in these areas. VPI&SU continuously promotes the transfer of this research to industry where it can be further developed into useful and state-of-the-art products.

Active control is most applicable to low frequencies where passive techniques may not be practical or limited. Impressive noise reductions have been demonstrated which would be difficult to achieve passively at low frequencies. The techniques developed in this program are attractive when structure-borne noise is a significant acoustic factor. The control approaches and structural actuators and sensors are considerably easier to implement compared to operating in the acoustic media with acoustic projectors and farfield sensors.

Another attribute is that active control can many times be retrofitted into existing structural systems where it is not practical to incorporate passive improvements for the same degree of performance. Passive approaches can involve major redesign efforts that may not be economical and practical. Active techniques are being developed which do not require the modification of existing systems and intrusion.

With the inclusion of electronic controllers these techniques have the unique advantage of being able to adapt to changing conditions. Such as time and environmental variability in coupling (structural and acoustic), and in the noise source mechanism.

While there are many attractive advantages to incorporating active structural acoustic control and adaptive material technologies there are some practical limitations. Most assuredly these technologies will increase the complexity of a structural quieting system which may also be costly to develop, produce, and maintain operationally. Also, active control is not a substitute for good passive design techniques, but it is complementary when properly integrated. For low frequency problems active techniques offer significant acoustic reductions at reduced weight and volume in comparison to the available passive options for the same degree of performance. An important requirement will always be designs which do not degrade the existing passive capability.

From a technical standpoint there are some difficult issues which must be addressed in the design and development of a practical active control system. While there are significant advantages at low frequencies the capabilities of the technology will diminish with increasing frequency. In these cases it has difficulty competing with available passive technologies. While most passive techniques are dissipative and stable in nature active control can present risks of instability and undesired gains in noise levels. Robust systems can operate under highly variable and adverse conditions, but typically with some loss in performance. The designs for specific disturbance sources and signatures to be controlled will more likely be unique in implementation, but offering significant performance.

Active structural acoustic control and adaptive/smart material technologies can have a significant impact upon underwater acoustics and submarine designs in the future. Sufficient basic research has been conducted to indicate that gains can be made in near-term applications such as: low frequency radiated noise, sonar self-noise, and interior air-borne noise. Also, it becomes apparent that to realize the optimum and full potential of these technologies they should be incorporated into the structural design rather than retrofitted into existing systems to cure design deficiencies. The U.S. Navy should advance and demonstrate active structural acoustic control and adaptive material technologies to the point that sound engineering tradeoffs can be implemented in the original design processes.

SECTION 9.0

FUTURE DIRECTIONS

In the future development of active structural acoustic control further refinements and advances can be made as well as extending the research to complex structures and more realistic design applications. This will require a highly coordinated and integrated technical effort as demonstrated during the three year period of this Grant.

Further studies are required in the optimal configuration of systems applied to structures with some degree of complexities, and which additionally offer a high degree of robustness. This requires a more in-depth examination of actuator and sensor placement, and the impact of a broad number of system design temporal and spatial errors. As the complexity of a structural system increases the importance of mode coupling will likewise increase. The complex issues of mode coupling and conversion need to be addressed and understood in terms of their impact upon active structural acoustic designs and approaches.

This research effort has demonstrated the potential of wavenumber domain techniques in the design of active structural acoustic control systems, and this work should be further developed. The implementation of active control into the NASHUA model should be continued such that large scale and realistic problems can be examined and evaluated in terms of the potential offered by a number of candidate active structural acoustic control approaches for fluid loaded structures.

The importance of sensing in the design of active structural acoustic systems has been demonstrated, and the unique benefits from optimally designed distributed sensors. Further work is required in the actual design of sensors and their use in a system. There is the unique requirement for the development of sensors which can be located near the structure that give estimates of far-field radiated acoustic pressure or power. A number of potential approaches need to be explored.

Of recent importance in fiber optic sensing has been the development of sensor fibers, which have weighted sensitivities along their lengths. The use of such fibers in structural vibration sensors allows the measurement of specific vibrational modes (radiating components) and the rejection of others (non-radiating components), permitting the implementation of simplified algorithms in structural acoustic control systems.

Adaptive materials and structures are gaining in importance with the wider use of basic composite structural materials, and further work is required in the development of new innovative designs of both actuators and sensors. More progress is required on the development of new theories and methods for active/adaptive shaped distributed actuators and sensors as applied to the more complex structures that are inherently offered by composite materials. Research needs to move in the direction of more complete and integrated intelligent structural,

and control system designs and architectures. This will require the integration of sensing, actuation, and signal processing functions within materials and structures.

This research effort has demonstrated the merits of the more conventional adaptive LMS and control theory approaches, and this work needs to be further advanced. In particular the adaptive LMS approach needs to be extended to broadband noise inputs, and the use of adaptive IIR filters. Hierarchical control approaches need to be pursued since they offer the potential of reduced system complexity and increased robustness in realistic and high order structural acoustic systems. Controller designs which exploit parallel processing hardware and hierarchical system strategies need to be developed and demonstrated. The required state-of-the-art in digital components and hardware is now available and affordable for these applications and problem areas.

The application of modern control theory has demonstrated its merits in the control of multidimensional structural acoustic systems (i.e., plants) and their acoustic radiation for arbitrary disturbances. This work needs to be expanded to address more complex structural systems, and the transition to more realistic situations and structures. This will require the development of higher performance systems in terms of dimensionality and functional data throughput (e.g., parallel and hierarchical control architectures). The design of control laws employed need to more tailored to the disturbance and true structural acoustic components of priority interest (i.e., in terms of spectral and radiation efficiency), and techniques such as H_∞ represent a move in that direction. There is the need for the development of reduced order modelling techniques while at the same time giving consideration to the robustness issues.

Model reference control approaches need to be more further investigated for both modern control theory and signal processing implementations. This will require the development of suitable models that reflect the desired structural acoustic and dynamic system behavior objectives.

In both the LMS feedforward and state feedback approaches the implications of total control system time delay and control system throughput requirements (e.g., sampling rates) need to be addressed for high percentage bandwidth (i.e., broadband) disturbances and related efficient structural acoustic responses. These issues can have a major impact on the basic control system sensor, actuator and controller system architectures and design approaches.

For the last five years VPI&SU has been engaged in the research and application of Artificial Neural Network (ANN) technology. Various configurations of ANN's can be trained with appropriate data or even numerical codes to give estimates of far-field pressure or intensity at multiple points from information arrays of accelerometers or microphones (or hydrophones) located on or near the surface. These techniques offer the potential of working with very complex structural shapes, disturbance sources (i.e., narrowband, broadband and transient), and in particular the presence of nonlinear mechanisms.

SECTION 10.0

CONCLUSIONS

The results presented here show that significant progress has been made towards both understanding the mechanisms of active structural acoustic control and ultimately implementing the technique in realistic situations. In conjunction with this, new understanding in the individual areas of distributed actuators and sensors as well as control theory and implementation as related to the structural acoustic problem has been achieved. The technique shows much potential for efficiently actively/adaptively controlling structure-borne noise radiation in many situations.

The key technical accomplishments under this Grant have advanced the state-of-the-art in structural acoustics, adaptive materials and structures, and active control. Concisely some of the important accomplishments are:

1. Demonstrated that structural acoustic control has a number of marked advantages over the use of acoustic secondary sources and sensors.
2. Applied active structural acoustic control to beam, plate and cylindrical structures.
3. Demonstrated active structural acoustic control of narrowband, wideband, and transient disturbances.
4. Developed, implemented and applied multi-input/multi-output (MIMO) adaptive LMS feedforward and closed loop state-space control theory designs.
5. Optimal active structural acoustic control approaches represent a significant reduction in the dimensionality of the control system compared to that of conventional active structural vibrational control.
6. Developed design approaches and algorithms for the optimal placement of point force and distributed PZT actuators for actively controlling acoustic radiation from plates.
7. Similarly, developed the design approaches for the optimal placement of point accelerometer, and distributed (attached and embedded) PVDF sensors.
8. Demonstration of optical fiber sensing for structural acoustic control of radiation.
9. Developed an eigen-analysis approach for designing shaped PVDF distributed structural acoustic sensors

10. Extended the capability of the DTRC NASHUA numerical structural acoustic model to address active control approaches for fluid loaded structures.
11. Analytical and experimental demonstrations and investigations of acoustic wavenumber control.
12. Demonstrated state-space control theory approaches which incorporate observers that estimate the acoustic farfield from structural measurements.
13. Development of a transputer-based parallel processing controller hardware system for multi-input multi-output (MIMO) experiments.
14. Demonstrations of embedded Shape Memory Alloy (SMA) fibers in composites for adaptive structural tuning to reduce acoustic radiation.
15. Implementation of embedded PVDF sensors and theoretical analysis of embedded PZT actuators in composites.
16. Extended basic feedforward adaptive control approaches and theory through an examination of the eigen-structure.
17. Technology transfer to defense industries for further development into state-of-the-art applications, and products.

SECTION 11.0

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APPENDIX A

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Shaw, J. K.	Prof.	Math

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Sudeora, S.	M.S.	EE
Thomas, D.	M.S.	EE
Tran, Tuan A.	M.S.	EE
Vengsarkar, Ashish M.	Ph.D.	EE
Vipperman, Jeff S.	M.S.	ME
Wang, Bor-Tsuen	Ph.D.	ME
Whittington, Doyle	M.S.	ME
Zhang, X. D.	Ph.D.	ME
Zhou, Ning	M.S.	ME
Zimmermann, B.	M.S.	EE

APPENDIX B

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STRUCTURAL ACOUSTICS

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